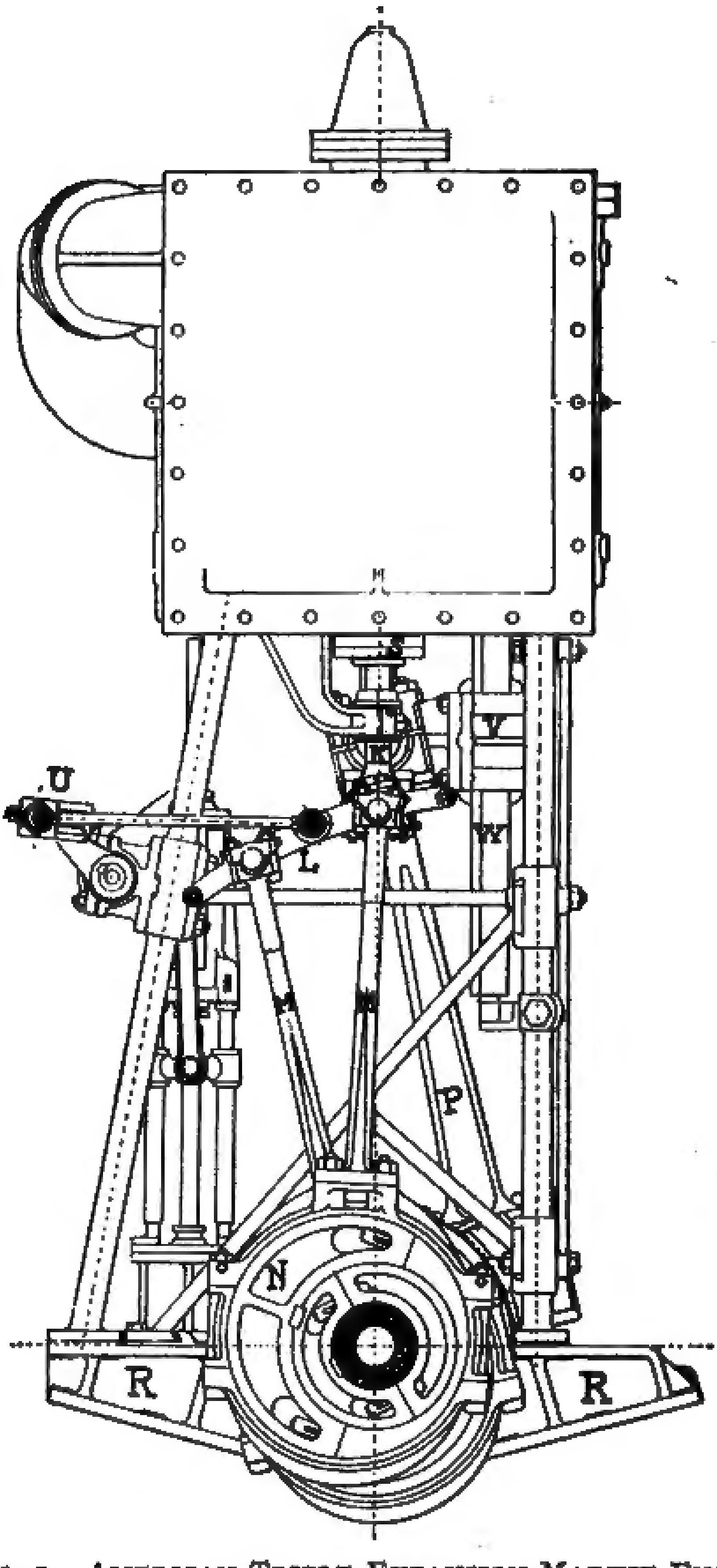

FIG. 1.—AMERICAN TRIPLE EXPANSION MARINE ENGINE.

FRONTISPIECE.

THE

AMERICAN MARINE ENGINEER,

THEORETICAL AND PRACTICAL.

WITH EXAMPLES OF THE LATEST AND MOST APPROVED AMERICAN PRACTICE.

FOR THE USE OF

MARINE ENGINEERS AND STUDENTS.

BY

EMORY EDWARDS,

MARINE ENGINEER,

ASSOCIATE EDITOR "THE AMERICAN SHIPBUILDER,"

Author of "A Catechism of the Marine Steam Engine," "Modern American Marine Engines, Boilers and Screw Propellers," "The Practical Steam Engineer's Guide," "Modern American Locomotive Engines," "The American Steam Engineer."

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This Volume is Dedicated

TO

ENGINEER-IN-CHIEF

GEO. W. MELVILLE, U. S. NAVY,

CHIEF OF THE BUREAU OF STEAM ENGINEERING, U. S. NAVY DEPARTMENT, WASHINGTON, D. C.

AN AMERICAN HERO,

Sans peur et sans reproche.

AS A TESTIMONIAL OF HIGH REGARD,

BY HIS FRIEND

THE AUTHOR.

PREFACE.

THIS work is presented to the Marine Engineers of the United States by one of their own number, who trusts that its use will be productive of satisfactory results.

The writer has endeavored to prepare a clear, concise, and thoroughly practical work; to treat each subject in as brief and concise a manner as possible, and yet preserve that clearness and fullness of statement so desirable, and even indispensable, in a text-book.

To secure this end, he has limited himself to statements of the essential facts and principles, excluding all unimportant details and avoiding all lengthy descriptions.

It was deemed advisable not to swell the volume to an inconvenient size by the insertion of obsolete matter, as it would only have increased its cost without increasing its value.

Marine Engineers should ever bear in mind that Marine Engineering is rapidly progressive, and that "we must, in this world, learn a good deal from the experiences of others. It will be a mighty small world that a man is introduced

to by his own experience." And therefore, if he wishes to "keep up with the procession," or, in other words, to keep himself posted as to the almost daily improvements that are being made in marine engines, boilers, etc., he must devote his leisure hours to *study*.

The writer is aware of the limited time any engineer has to read and study. His daily duties frequently draw heavily on his physical and mental strength, and if he is to learn about what is going on in his own special line of occupation, he has to have the information presented in as clear and as comprehensive a manner as possible. If he has to study up something first, before he can understand what is written for his special benefit, very likely it will not receive any attention, but be cast to one side as too difficult for him to understand, and requiring too much time to unravel its meaning.

Besides the vast advantages which a sufficient knowledge of mathematics and a facility in drawing give the intelligent inventive engineer over the mere workman, the former generally has another advantage, almost as great, in knowing what has been done in the world. While the unlearned engineer knows only what has been done in his neighborhood, and despises whatever is in books or mechanical journals, the intelligent engineer, by books and journals, learns what has been done in all countries that

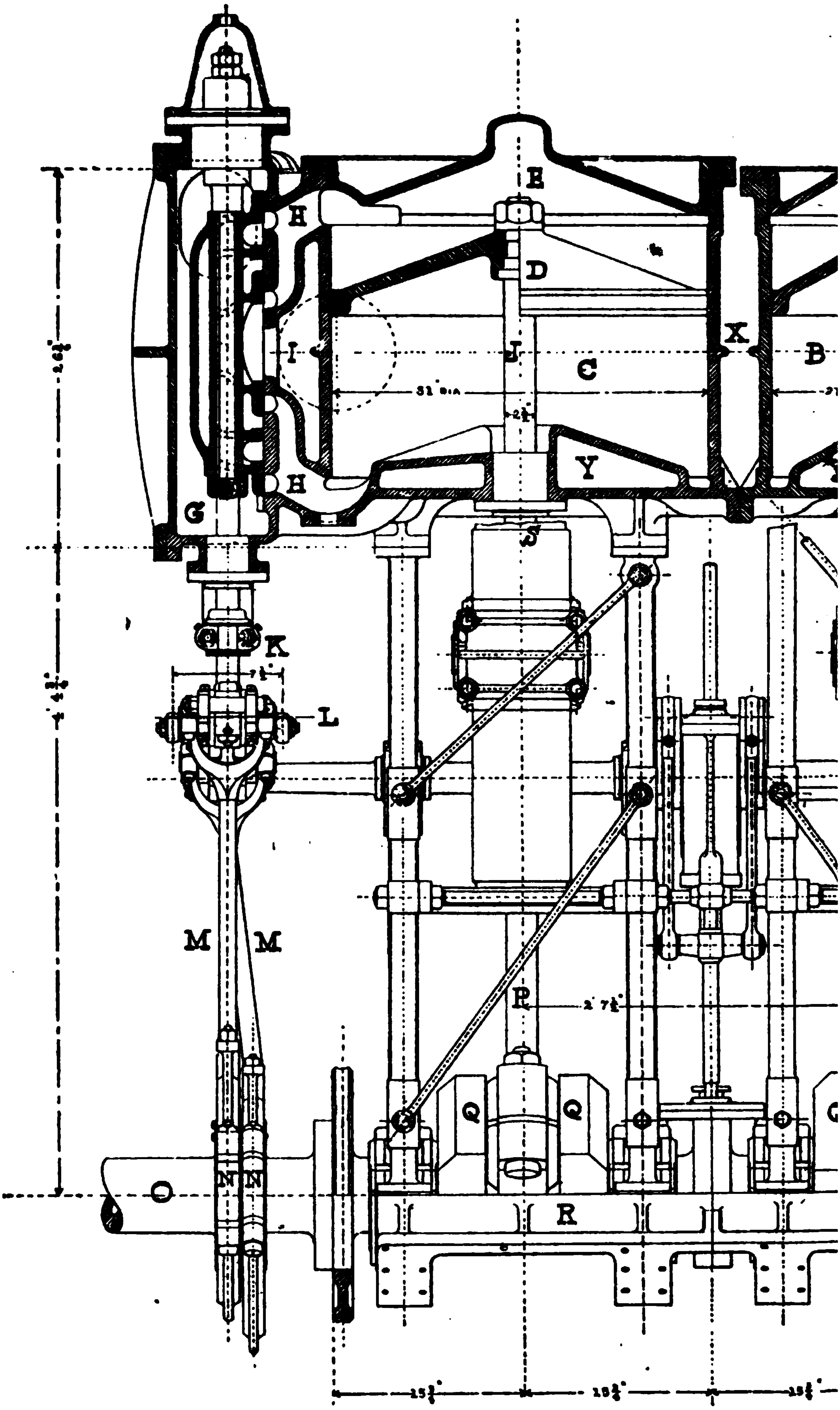

FIG. 2.—AMERICAN TRIPLE E

:PANSION MARINE ENGINE.

OPPOSITE PAGE [illegible].

are so far civilized as to have engineering publications; and when he is planning new work, he has in his mind all the resources of the world, in addition to the scanty resources of local practice.

Practice, to a great extent, at least great enough to understand it, is necessary for an engineer; but practice to this extent is soon acquired; a young man from a good school or college, where he has been well grounded in theory, will in a year's apprenticeship, under a good foreman, become a better engineer than if he had spent in the shop the time he spent in the schools. Practice can be acquired at any time; but theory, if not acquired at the outset, will rarely uproot the bad mental habits that grow up in uncultivated minds.

It is gratifying to know of the efforts to gain knowledge that are now being made in the assembly rooms of the numerous councils of the M. E. B. A., and it is also gratifying to learn of the appreciation with which engineering journals are now received, not only by the constructing engineers, but by the intelligent, energetic and practical Marine Engineers who carefully peruse them.

Of the many American publications devoted to steam engineering now published in this country, "The American Machinist," "The Engineer," and "The American Shipbuilder,"

all of New York; "The American Engineer," of Chicago; "The Western Machinist," of Cleveland, O.; and "Mechanics," Philadelphia, Pa., are worthy of the support of every American Marine Engineer who wishes to keep in touch with the times.

The writer herewith tenders his best thanks to his friends, Prof. T. M. Goodeve; A. E. Seeton, M. I. N. A., the author of that most masterly work "A Manual of Marine Engineering;" Engineer-in-Chief Geo. W. Melville, U. S. Navy; Prof. R. H. Thurston, Ph.D.; Frank C. Smith, M. E.; Jos. R. Oldham, N. A.; Prof. J. A. Whitman, M. E.; Prof. W. Thorn, M. E. N. A.; the "Journal of Commerce," of Boston, and also to the enterprising publishers, for much valuable data and assistance in the preparation of these pages.

As this book is intended as a supplement to his "Catechism of the Marine Steam Engine," the writer desires to call the particular attention of young Marine Engineers and firemen to that work.

THE AUTHOR.

Baltimore, Md., June 15th, 1891.

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INTRODUCTION.

As this book may fall into the hands of some young beginner or student who is taking his first lessons in steam engineering, the Author believes it advisable to devote this Introduction to a little elementary instruction for the purpose of aiding such a learner to grapple successfully with the succeeding chapters.

For this purpose he gives two working drawings of a modern American triple expansion marine engine, showing the various parts with all necessary fulness, with the dimensions of the principal ones. (See Figs. 1 and 2.)

A shows the High Pressure Cylinder.
B " Intermediate Pressure Cylinder.
C " Low Pressure Cylinder.
D " Pistons.
E " Cylinder Heads or Covers.
F " Piston Valves of the H. P. and I. P. Cylinders.
G " Slide Valve of the L. P. Cylinder.
H " Steam Ports.
I " Exhaust Ports.
J " Piston Rods.

K shows the Valve stems.
L " Link.
M " Eccentric Rods.
N " Eccentrics.
O " Portion of Shaft.
P " Connecting Rods.
Q " Cranks.
R " Bed Plate.
S " Stuffing Boxes.
T " Steam Chest.
U " Starting Gear.
V " Guide Block and Cross Head.
W " Slide.
X " Receivers.
Y " Cylinder Bottom.
Z " Pillow Blocks and Brasses.

The student will find the uses of the various parts illustrated and described in the succeeding chapters.

Nearly all beginners and not a few of the older engineers who have had no experience with triple expansion engines either in their construction or management, seem to think there is something mysterious about them which only an expert can properly understand. Now if the student will look at a triple expansion as simply a combination of three separate and independent engines working side by side, the difficulty will vanish at once.

The operation of the steam is as follows: The

steam from the boilers having been admitted by the valve to the High Pressure Cylinder and done its work, instead of being exhausted into the atmosphere as in case of a non-condensing engine, or at once into the condenser as in the case of a single cylinder condensing engine—passes into a space between the High Pressure Cylinder and the Intermediate Cylinder. This space is called a "Receiver," and acts as a steam reservoir for the Intermediate Cylinder, and from which it gets its steam. The steam is now at a greatly reduced pressure, and the diameter of the I. P. Cylinder is correspondingly increased to compensate for the reduced pressure. The steam now passes into the Intermediate Cylinder, does its work there, and is in turn exhausted into another Receiver, which is between it and the Low Pressure Cylinder. Again the steam is at a much greater reduced pressure, and it will be noticed that the diameter of the L. P. Cylinder is greatly increased to make up for this loss of pressure. From this second Receiver the steam enters the Low Pressure Cylinder, does its work there, and is then in turn exhausted into the condenser in which it is condensed (turned into water again) which forms a vacuum in the condenser and on the exhaust side of the low pressure piston. By this means the atmospheric pressure is got rid of, and as a reduction of pressure on one side is equivalent

to an increase of pressure on the other, we gain about 12 pounds pressure, in practice—the vacuum not being a perfect one: about 28 inches.

The air-pump removes the condensing water, with its uncondensed vapor, air—and keeps a partial vacuum constantly in the condenser while the engine is in operation.

CHAPTER I.

THE THEORY OF THE STEAM ENGINE.

FOR many years engineers cared nothing about the theory of the steam engine. They went on improving and developing it without any assistance from men of pure science. Indeed it may be said with truth that the greatest improvement ever effected—the introduction of the compound engine—was made in spite of the physicist, who always asserted that nothing in the way of economy of fuel was to be gained by having two cylinders instead of one. In like manner, the mathematical theorist was content to make certain thermo-dynamic assumptions, and, reasoning from them, to, construct a theory of the steam engine, without troubling his head to consider whether his theory was or was not consistent with practice. Within the last few years, however, the theorist and the engineer have come a good deal into contact, and the former begins at last to see that the theory of the steam engine as laid down by Rankine, Clausius, and other writers, must be deeply modified, if not entirely re-written, before it can be made to apply in practice. We have recently shown what

M. Hirn, who combines in himself practical and theoretical knowledge in an unusual degree, has had to say concerning the received theory of the steam engine, and its utter inutility for practical purposes; and papers recently read before the Institutions of Mechanical and Civil Engineers, and the discussions which followed them, have done something to convince mathematicians that they have a good deal yet to learn about the laws which determine the efficiency of a steam engine. It has always been the custom to class the steam engine with other heat engines. It is now known that nothing can be more erroneous. The steam engine is a heat engine *sui generis*, and to confound it with a hot-air engine, or any motor working with a non-condensible fluid, is a grave mistake. It is not too much to say that many engineers now understand the mathematical theory of the steam engine better than do men making thermodynamics a special study. But there remains a large number of engineers who do not as yet quite see their way out of certain things which puzzle them, or which they fail to understand. There are, indeed, phenomena attending the use of steam which are not yet quite comprehended by any one, and we may be excused if we say something about one or two points which require elucidation.

One of these is the mode of operation of the

steam jacket. It is a very crude statement that it does good because it keeps the cylinder hot. It might keep the cylinder hot, and yet be a source of loss rather than gain; and, as a matter of fact, it is doubtful now if the application of steam jackets to all the cylinders of a compound engine is advisable. It is well known, too, that circumstances may arise under which the jacket is powerless for good. Thus, for example, the late Mr. Alfred Barrett, when manager of the Reading Iron Works, carried out a very interesting series of experiments with a horizontal engine, in order to test the value of the jacket. This engine had a single cylinder fitted with a very thin wrought-iron liner, between which and the cylinder was the jacket space. The jacket was very carefully drained, and could be used either with steam or air in it. Experiments were made on the brake with and without steam in the jacket. The result was a practically infinitesimal gain by using steam in the jacket. In one word, the loss by condensation was transferred from the cylinder to the jacket. On the other hand, it is well known that single cylinder condensing engines must be steam jacketed if they are to be fairly economical. Circumstances alter cases, and the circumstances which attend the use of jackets are more complex than appears at first sight.

In considering the nature of the work to be

done, we must repeat a fundamental truth which was first enunciated by "The Engineer" (English). A steam engine can discharge no water from it which it did not receive as water, save the small quantity which results from loss by external radiation and conduction from the cylinder, and from the performance of work. At first sight the proposition looks as though it were untrue. Its accuracy will, however, become clear when it is carefully considered. After the engine has become fully warmed up, the cycle of events is this: Steam is admitted to the cylinder from the boiler. A portion of this is condensed. It parts with its heat to the metal with which it is in contact. The piston makes its stroke, and the pressure falls. The water mixed with the steam is then too hot for the pressure. It boils and produces steam, raising the toe of the diagram in a way well understood and needing no explanation here. During the return stroke the pressure falls to its lowest point, and the water being again too hot for the pressure, boils, and is converted into steam, which escapes to the atmosphere or condenser without doing work, and is wasted. The metal of the cylinder, etc., falls to the same temperature as the water. At the next stroke the entering steam finds cool metal to come into contact with, and is condensed, as we have said, and so on. But the quantity condensed during the steam

stroke is precisely equal to that evaporated during the exhaust stroke, and consequently no condensed steam can leave the engine as water.

Let us suppose for the sake of argument, however, that an engine using 20 lbs. of 100 lbs. steam per horse per hour discharges 2 lbs. of water per horse per hour. As each of these brought, in round numbers, 1185 thermal units into the engine and takes away only 212 units, it is clear that each pound must leave behind it 973 units; consequently the cylinder will be hotter at the end of each revolution than it was at the beginning, and the process would go on until condensation must entirely cease. It will be urged, however, that a steam jacket certainly does discharge water, and that in considerable quantity, which it did not receive; and as this is apparently indisputable, we are here face to face with one of the puzzles to which we have referred. The fact, however, is in no wise inconsistent with that advanced. If an engine with an unjacketed cylinder regularly receives water from the boiler, that engine will discharge precisely an equal weight of water. The liquid will pass away in suspension in the exhaust steam. The engine has no power whatever of converting it into steam. The case of a jacketed engine is different. Such an engine will evaporate in the cylinder water received with the steam, but it can only do so at the expense

of the steam contained in the jacket. For every 1 lb. of water boiled away in the cylinder 1 lb. of steam is condensed in the jacket; and the corollary is that if an engine was supplied with perfectly dry steam there would be no steam condensed in the jacket, save that required to meet the loss due to radiation and the conversion of heat into work. The effect of the jacket will be to boil a portion of the water during the close of the stroke, and so to keep up the toe of the diagram, and so get more work out of the steam. If, however, the steam was delivered wet to the engine, it is very doubtful if the jacket could be productive of much economy. The water would be converted into steam during the exhaust stroke, and no equivalent would be obtained for the steam lost in the jacket.

In a good condensing engine, about 3 lbs. of steam per horse per hour are condensed in the jacket. The cylinder will use, say, 15 lbs. of steam, so the total consumption is 18 lbs. per horse per hour. It is none the less a fact, although it is not generally known, that the average marine boiler sends over about 8 per cent. of water in the form of insensible priming with the steam. Now, 8 per cent. of 18 lbs. is 1.44 lbs., so that in this way we have nearly one-half the jacket condensation accounted for as just explained. One horse-power represents 2562 thermal units expended per hour, or say

2.6 lbs. of steam of 100 lbs. pressure condensed to less than atmospheric pressure; and 1.44+2.60 =4.04 lbs. per horse per hour, as the necessary jacket condensation if no water is to be found in the working cylinder at the end of each stroke. That this quantity is not condensed only proves that the water received from the boiler, or resulting from the performance of work, is not all re-evaporated.

Something still remains to be written about the true action of the steam jacket, but this we must reserve for another chapter. We have said enough, we think, to show that, as we have stated, the jacket has more to do than to keep the cylinder hot. With jacketed engines, more than any other, it is essential that the the steam should be dry. In the case of an unjacketed engine, water supplied from the boiler will pass through the engine as water, and do little harm; but if the engine is jacketed, then the whole, or a part of this water, will be converted into steam, especially during the period of exhaust, when it can do no more good than if it were boiled away in a pot in the engine room. This is the principal reason why such conflicting opinions are expressed concerning the value of jackets. That depends principally on the merits of the boiler.

COMPOUND, TRIPLE EXPANSION AND QUADRUPLE EXPANSION ENGINES.

Cylinder Condensation and Re-evaporation.—The method of variation of this waste was qualitatively determined by Clarke about 1850; was roughly gauged by him, both as to magnitude and as to its effect in limiting the ratio of expansion; was quantitatively investigated by Hirn and Isherwood afterward, and was finally made the subject of an investigation by Messrs. Gately and Kletsch, in which it was endeavored to ascertain with some degree of accuracy the method of the variation of the waste with variation of each of the essential conditions affecting and determining it. The result of the research in brief was to show that the waste varied, in the cases studied, sensibly as the square root of the expansion and as the time of exposure, and was subject to a very slow decrease as the pressure adopted increased, the engine being worked condensing; decreasing about twice as rapidly, the condenser being thrown off.

Variation with ratio of expansion was also capable of being expressed with great accuracy by an hyperbolic expression, the product of areas of surface exposed up to the point of cut-off and the percentage of condensation being found sensibly constant. Under ordinary working conditions, the steam pressure being about sixty pounds per square inch, by gauge, the cut off at

one-third, and the speed of piston 554 feet per minute, and of rotation sixty-eight revolutions, the condensation was about one-third, or the equivalent of fifty per cent. of the total consumption of a similar engine having a non-conducting cylinder, and thus free from this waste. Reduced to quantity of steam and of heat wasted, per square foot of surface exposed to point of cut off, per minute of exposure, and per degree of range of temperature between prime and exhaust steam, Professor Marks finds the co-efficient to be .02 pounds, or 18 B. T. U., nearly, a result closely confirmed by the investigations of the same author, taking Hill's experiments for comparison, and also corroborated by the later work of other investigators.

These facts and laws being established, it becomes possible to determine the behavior of steam entering any given cylinder, and its method of working and of waste in any engine. Common experience, as well as theoretical considerations based upon the investigations already made, proves that it is impossible to expand steam in the ordinary single-cylinder engine with satisfactory gain of efficiency, beyond a point variable with the conditions assumed, but which may be roughly taken as not far from that giving a ratio of expansion equal to about one-half the square root of the steam pressure, measured from vacuum for the condensing engine

of common type, or as measured by gauge for the non-condensing engine. The waste by internal condensation increasing as the point of cut off is shortened up, the loss, after a time, compensates the gain by increased expansion, and a point of maximum economy is passed at a very early stage for the older types of engine and later, but still at a comparatively low value of the ratio of expansion, for modern engines. For example, the old beam engine, such as was and is still used on American rivers, or in the lake trade, with steam at twenty-five pounds by gauge and a low piston speed, had a ratio of initial to back pressure plus friction of about 8 to 1; but its best ratio of expansion for efficiency of fluid was about 2 or 2½. The same type of engine, with twice the pressure per gauge, has values of these two ratios of about 16 and 4 or 5 respectively. The ratio of expansion for maximum efficiency of working fluid was thus but about one-fourth that which thermo-dynamic theory unqualified, would dictate. As engines have been improved this discrepancy has been reduced, but it still remains, with the best of engines, considerable.

The amelioration of waste thus becomes an important matter. It thus happens that the efficiency and economy of operation of the single cylinder, the "simple" engine, is at all times limited by this very serious internal waste; and

the question which all engineers since Watt have been endeavoring to solve, is: In what manner may we best proceed to eliminate or ameliorate this loss? Three methods which have been found advantageous, and, in special cases, fairly effective, are:

(1) Superheating;
(2) Steam jacketing;
(3) "Compounding."

It is evident that, if the steam can be introduced into the engine at such a temperature that the cooling action of the metal of the cylinder will not cause its condensation initially, and the stroke may be performed without condensation in consequence of doing work, no loss of heat from the cylinder can take place by re-evaporation; and if no such loss occurs, the waste of heat at entrance, in turn, by initial cooling, will be reduced. Super-heated steam, also, is a non-conductor and a non-absorbent of heat, precisely like the permanent gases. It is thus, also, less liable to waste. But it is found in practice that superheating beyond a moderate degree, perhaps 100 degrees Fahrenheit, is inadvisable on account of risks to engines, and cost of repairs to superheater, which more than compensate its advantages. It has come to be regarded as an auxiliary in economizing, not as a remedy for interior wastes.

Steam jacketing is another and common par-

tial remedy for the waste. By surrounding the steam cylinder with the steam jacket, it is possible to produce, in part, the effect of superheating; that is, to secure dryer steam in the engine throughout the stroke. The amount of re-evaporation, during the period succeeding cut off and up to the closure of the exhaust valve, and the quantity of heat of which the cylinder is thus robbed, measures the amount of initial condensation and waste, and the weight of steam which must be supplied in excess of the thermo-dynamic demand to compensate that loss. The effect of the addition of a steam jacket depends upon the conditions of operation of the engines, largely, and may be productive of marked advantage, or, under unfavorable conditions, of no important useful effect. With steam initially dry or superheated, the jacket is probably always decidedly helpful; but with wet steam it is of comparatively little value, even if not sometimes a positively wasteful adjunct. High-speed engines derive less advantage from its application than slow-moving engines; and compound or multiple cylinder engines are less dependent upon it for economy than are simple engines. The saving effected in ordinary cases, by its use, may be taken as averaging about 20 per cent.; and about the same gain is attained by effective superheating within the usually practicable range. The two devices in conjunction may be

expected ordinarily in marine engines, perhaps, to give a gain of something like 30 per cent. as compared with the standard forms of unjacketed simple engine working with slightly wet steam. The addition of either expedient to the latter practice, if properly performed, considerably increases the magnitude of the ratio of expansion at maximum efficiency of fluid. Where it would ordinarily be approximately equal to one-half the square root of the pressure, as above, it might become, with superheating or with steam jacketing, a figure as much as 30 or 40 per cent. higher; and both expedients together might nearly double the profitable ratio of expansion. The assumption is commonly made that the superheating is retained throughout the stroke and that steam jacketing may be relied upon to keep the working charge dry and saturated throughout the stroke; but neither of these hypotheses, as employed in the theory of the engines, is practically correct.

"Compounding," or the use of the multi-cylinder engine, in which the steam exhausted from one cylinder is again worked in a succeeding one, is the most familiar of devices for extending the economical range of expansion and increasing the efficiency of the engine. The limit to the useful extension of the expansion of steam in a single cylinder is found to be determined by the magnitude of the wastes in-

curred in the operation of an engine of which the working cylinder is a good conducting material. Any method of reducing this waste of heat internally will enable the efficiency of the engine to be increased by further profitable extension of the ratio of expansion. Common experience with the best constructions, and considerations which need not be here reviewed, shows that the engineer may reasonably expect, by good design, construction and management, to secure an economy of steam which is fairly measured by the following table, the ratios of expansion, *r*, taken being, for each case, those which give best results for a given engine:

STEAM PER HORSE-POWER PER HOUR AT LEAST RATIOS OF EXPANSION IN BEST ENGINES.

r	3	4	5	6	7	8	10	12	15	20	25	50	75
Lbs.	32	27	25	22	20	20	19	17	16	15	15	1.1	0.9
Kgs.	15	12	11	11	9	9	9	8	7	7	7	05	04

and ten per cent. better figures than these have been actually reported in peculiarly favorable cases.

Assuming it to be possible to divide the waste by cylinder condensation and leakage by two or more, it is evident that the limit to economical expansion and transformation of heat into work will be set correspondingly further away. This is precisely what is done by the multi-cylinder engine. The internal wastes are reduced approximately to those of a single cylinder, and

the gross percentage of waste is made less in the proportion of this division. The heat and steam rejected as waste by internal transfer without transformation from the first cylinder, are utilized in the second nearly as effectively as if they were received directly from a boiler at the pressure of rejection from the first cylinder. In so much, therefore, as the pressure can be increased and the increase utilized by the addition of another cylinder, gain is secured. If the total ratio of expansion can thus be raised, under the best working conditions for each case, from, we will say, four up to eight, we should hope to secure a reduction of coal consumed from two and a half, we will say, to two pounds per horse power and per hour, which is about the average figure in good practice.

The practical questions thus meet the engineer: To what extent can this principle be availed of? What range of pressure and what ratio of expansion should be assigned to a single cylinder? and how many cylinders should be adopted to give the best results with the highest steam pressure practicable for a specified case? Common experience aids in solving this problem, by showing that the very best results are ordinarily obtained, in each class of multi-cylinder engines, when, the engine being properly designed for its work, terminal pressure for the system can be economically made something

above the sum of back pressure in the low-pressure cylinder, plus friction of engine, This total may usually be taken probably at eight or ten pounds above vacuum. The latter figure will be here assumed.

The fundamental principles are now easily perceived. There are three main facts upon which to base our theory of the multi-cylinder engine. These are:

(1) Economical expansion in a single cylinder has a limit due to increasing internal wastes, which is found at a comparatively low ratio of expansion.

(2) The method of expansion may be, for practical purposes, such as are here in view, taken to be approximately hyperbolic; the terminal pressure being somewhat above that which corresponds to the sum of all useless resistances, and which may be here taken, as for example, about ten pounds per square inch above vacuum. The division of the initial pressure by this terminal pressure will thus give an approximate measure of the desirable ratio of total expansion for the best existing engines.

(3) All steam entering any one cylinder will be rejected, as steam, into the succeeding cylinder, external wastes being neglected, and into the condenser; and the full amount of steam condensed at entrance by absorption of heat by the interior surfaces of the cylinder will be re-

evaporated later, and will pass into the condenser or into the next cylinder, and heat transferred in the one direction, in the one process, will be transferred in precisely equal amonnt in the opposite direction in the other.

The last point is a very important one, and it is very easily established. The cylinder, when in steady operation, is neither permanently heated nor permanently cooled; no progressive heating can go on, as it would, in that case, become heated above the temperature of the steam and become a super-heater; no progressive cooling can occur, since in that case the cylinder would become a condenser of indefinite capacity. It must, therefore, transfer to the next element of the system all the heat which it receives, assuming that external radiation and conduction may be neglected, and that the Rankine and Clausius phenomenon of internal condensation, by transformation of heat into work, is ignored. It also further follows that the introduction of one or of many cylinders between the terminal element and the boiler does not, through cylinder condensation, affect the operation of the latter cylinder, however great that condensation may be, provided the operation of the added elements is effected by raising the steam pressure commensurately, leaving the final element of the series the same initial pressure as before. The total waste by this form of

loss is thus evidently measured, in the case of the multi-cylinder engine, by the maximum waste in any one cylinder. If all are equally subject to this loss, the rejected steam of re-evaporation from any one cylinder, supplies precisely what is needed to meet the waste by initial condensation in the next; and so on through the series. Thus the use of a series of cylinders, in this manner, divides the total waste for a single cylinder, approximately at least, by the number of cylinders; and it is in this manner that the compound system gives its remarkable increase of efficiency. As stated by the writer,* many years ago, "The serious losses arising from condensation and re-evaporation within the cylinder, and which place an early limit to the benefit derivable from expansion, affect both types of engine, and so far as seems now known, equally;" but the modern type permits the interception of the heat wasted from one cylinder, for utilization by its successor, in such manner that the total waste becomes, practically, that of the low pressure cylinder alone. If any one cylinder wastes more than another, the total waste is, as above stated, measured more nearly by the loss in the most wasteful member of the system.

Thus the three principles which have been

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* Thurston.

above enunciated give a means of constructing a philosophy of the multi-cylinder engine, which will meet the essential needs of the designer and of the student of its theory. The first principle shows that a limit existing to economical expansion in a single cylinder, the advisable number of cylinders in series may be determined, when that limit is ascertained, either by experiment, by general experience, or by rational theory and computation. The second principle shows that we may find a tentative measure, at least, of the desirable ratio of expansion for maximum density when the best terminal pressure for the chosen type of engine is settled upon. This total range is divided by the admissible range for a single cylinder, raised to a power denoted by the number of cylinders. Combining thus the two considerations referred to, we obtain a determination, probably fairly approximate, of the proper number of cylinders in series. The third principle permits an estimate to be made of the probable internal wastes of the series, and the probable total expenditure of heat and of steam, and a solution of all problems of efficiency for the compound engine of whatever type.

The first step in the process is evidently the determination of the best ratio of expansion, under the assumed condition of operation and for the given type of engine, for a single cylin-

der; then the best ratio of expansion for the series, all things considered, this study being made from the financial standpoint, as must be every problem which the engineer is called upon to solve. It is not the thermo-dynamic, nor the fluid, nor even the engine efficiency, which must be finally allowed to fix the best ratio of expansion; but it must be the ratio of expansion at maximum commercial efficiency, that which will make the cost of operation at the desired power a minimum for the life of the system. The total ratio being settled upon, and that allowable as a maximum, for a single cylinder, it is at once easy to determine the best number of cylinders in series. The first-mentioned ratio is that at maximum commercial efficiency, as just stated; but the second must be taken as that which gives the highest efficiency of engine, the back pressure in that cylinder, and the friction of the cylinder taken singly, being considered, together with its proper proportion of the friction of the engine as a whole.

Studying the method of distribution of waste among the several cylinders of the multi-cylinder engine, it will be observed that, since the pressures increase more rapidly than the temperatures, the range of temperature in the high pressure cylinder is greatest; while, the same weight of steam passing through the whole series, the low-pressure cylinder presents the

largest area of condensing surface in proportion to steam used. These differences are to a certain extent, though not wholly, compensatory. It may be assumed, however, without serious error, that the necessity of applying jackets or other methods of reducing internal wastes, will apply substantially as imperatively to one cylinder as to another; and that the adoption of a common ratio of expansion for both or all cylinders, or of apportioning the rates with reference to the equal division of power among them, will be found perfectly admissible, and will introduce no serious avoidable loss. Authorities have greatly differed in their views as to the relative advantage of jacketing one or another cylinder; but it is at least safe to jacket all, and probably, as above indicated, best to do so. The importance of the jacket evidently becomes less as other expedients for reducing wastes of this kind are adopted and are made more effective; as by increasing speed of engine, by superheating, by reheating between cylinders; and cases may be imagined in which the jackets may cease to have sufficient value to justify the acceptance of the risks and expense incurred in their employment. The same is true of the more complicated forms of valve gear needed to secure an approximation to the ideal distribution of steam,

CHAPTER II.

EXPANSION OF STEAM.

IT is well known that the amount of work which can be obtained from a given weight of steam, when not used expansively, is practically independent of the pressure at which the steam is worked. Thus, if steam of 100 pounds per square inch absolute pressure be admitted behind a piston one square foot in area, then by the time the piston has been pushed through 4.38 feet, exactly one pound of steam has been admitted, and the work done by it during admission is equal to the pressure on the piston multiplied by the number of feet through which the piston moves, or $100 \times 144 \times 4.38 = 63{,}072$ foot pounds. If, then, the steam escapes into the air, this represents the total work done by the steam.

Suppose now, that the steam used had been at a pressure of twenty pounds per square inch, and that this steam had been admitted behind a piston one square foot in area, then by the time the piston had been pushed through 19.7 feet, exactly one pound of steam at twenty pounds absolute measure has been admitted, and the work done by it during admission is equal to

$20 \times 144 \times 19.7 = 56,736$ foot pounds. But the pound of steam at 100 pounds pressure did 63,-072 foot pounds, which is not much greater than that done by the pound of steam at twenty pounds pressure. No advantage, however, has here been taken of the expansive power of steam, which increases as its initial pressure increases. Suppose we wish the terminal pressure in the cylinder to be twenty pounds absolute, then evidently the 1 pound of steam at 20 pounds cannot do any more work on the piston than is given above, namely, 56,736 foot pounds. On the other hand, the pound of steam at 100 pounds pressure can do a great deal more work, if, instead of exhausting into the air at that pressure, it is allowed to push the piston forward till it expands and becomes reduced in pressure to twenty pounds. This capacity for doing more work must have struck the most careless observer when standing by the exhaust pipe of a high-pressure engine.

Now the steam at 100 pounds pressure pushed the piston through 4.38 feet and escaped into the air, but it would push the piston through five times this distance in expanding to a terminal pressure of 20 pounds, or $4.38 \times 5 = 21.90$ feet, which is nearly the same distance as the 1 pound of steam at 20 pounds pushed the piston, namely, 19.7 feet. The piston in the two cases has been pushed through about the same dis-

tance by the same weight of steam; but observe the difference in the amount of work done in the two cases. In the first case the mean pressure on the piston varies from 100 pounds per square inch at one end of the stroke to 20 pounds per square inch at the other end, giving a mean pressure throughout the stroke of 52.2 pounds. In the second case the pressure on the piston was 20 pounds throughout. Hence the gain by using the high-pressure steam is $52.2 \div 20 = 2.61$; in other words, 2.61 times the amount of work is done by using the high-pressure steam expansively. The significance of this result becomes more apparent when we remember that the 1 pound of steam at 100 pounds pressure only costs an appreciably small quantity of fuel more for its production than the 1 pound of steam at 20 pounds—so small, in fact, that it may be neglected in practice.

To take another view of the case: Suppose the steam at 100 pounds pressure, instead of having been cut off at one-fifth of the stroke, and expanded to twenty pounds pressure as before, had been supplied to the cylinder at 100 pounds pressure throughout the whole stroke, then the mean pressure would be 100 pounds per square inch, and amount of work done would be five times as great as when steam at 20 pounds was admitted throughout the whole strol-e; but it would only be $5 \div 2.61$, or say twice as great as

the work done by steam admitted at 100 pounds and cut off at one-fifth of the stroke. In other words, when we admit steam at 100 pounds throughout the whole stroke, we use five times the weight of steam, and, therefore, also five times the weight of fuel, and only obtain twice the amount of work. Hence the economical advantage of using steam expansively. No account has been taken of the effect of back pressure.

To illustrate the advantage of expansive working, we will take an actual case from practice. A steamer of 1000 indicated horse power, having a pair of two-cylinder compound oscillating paddle-wheel engines of modern construction, runs at the following speeds and coal consumption for varying degrees of cut-off in each cylinder:

Point of cut-off.	Knots per hour.	Coal Consumption. pr. hr. in cwts.	Coal Consumption. pr. hr. in cwts.
3-10ths	8	6	0.75
4-10ths	9	9	1.00
5-10ths	10	12	1.20
6-10ths	12	20	1.66

From this table it will be seen how the weight of steam supplied to the cylinder affects the speed and coal consumption. Comparing the effects of the two extreme points of cut-off, when cutting at six-tenths instead of three-tenths, twice the volume and weight of steam is used,

3.3 times the weight of coal is consumed per hour, and 2.2 times the weight of coal is consumed per mile for 1.5 times the number of revolutions.

The effect of expansive working on the possible distance which a vessel can run with a given weight of fuel will also be evident; for in the case we are considering the vessel would run 2.2 times the distance when cutting off at three-tenths that she would run when cutting off at six-tenths. The influence of this increased economy of expansive working on the power to run longer voyages where coal is not easily obtained, has had immense influence on commerce with distant parts of the globe.

CHAPTER III.

THE EFFICIENCY OF STEAM JACKETS.

THE relative value of a jacketed or non-jacketed cylinder of an engine is a subject of much discussion. Tests are usually made on an engine at one time using the jacket and at another not using it, a condition somewhat against the jacket, as the jacket itself performs in a minor degree the purpose for which it was added even when steam is not admitted to it. English accounts are given of elaborate experiments with a triple expansion engine at the Whitworth Engineering Laboratory, Manchester, England, by which it was shown that steam jackets have a certain value.

The engines were throughout specially designed and constructed with a view to the fulfillment of the requirements. There were three separate inverted cylinder engines working on separate brakes, having the following dimensions:

Engine.	Cylinder. Diam. Inches.	Crank-shaft. Stroke. Inches.	Crank-shaft. Diam. Inches.
No. I (high-pressure)	5	10	2¾
No. II (intermediate)	8	10	2¾
No. III (low-pressure)	12	15	4
Air pump on No. III	9	4½	
Feed pump on No. III	1½	2	

All the engines had their cyl
both covers separately jacketed,
that they could be worked with c
in any or in all of the jackets. I
designed to carry a pressure ur
on the square inch, to run at ai
up to 1000 feet per minute, and
gear to cut off from zero up t
stroke. One engine was furnis
face condenser having 160 squar
the other two engines were furni
nate exhausts, either into the atı
steam-jacketed receivers supplyi
gine, each receiver having an a
of steam from the boiler. The
the locomotive type, with five
grate, 200 square feet of heati
carried 200 pounds of steam p
It was set in a hot chamber, wit
having 50 square feet of heat
which an area of 40 square feet
by scrapers, and was so arrange
moved in the opposite directio
The furnace was worked either
draught or with forced draught
stokehold system, and burnt
pounds an hour. The specialt
consisted, mainly, in the prov
render possible the accurate d
the manner in which each pa

work, as well as to make the performances of each organ, as far as practicable, independent of the performance of the rest. To accomplish this, the boiler and three engines had been separated by intervals of 20 feet, 7 feet, and 12 feet, and the steam distributed by five systems of pipes, while the engine shafting extended over a length of 36 feet. This spreading out of the engines entailed greatly increased radiation and additional friction. These quantities, however, being differently measurable, did not confuse the results.

The investigation, commenced in March, 1888, had been continued at the rate of two trials a week during the session, a trial occupying six, four or two hours, and being conducted as regular work in the Laboratory. As the trials were intended eventually to cover all possible systems of using steam, each system was fully investigated in a series of trials, giving consistent results, before proceeding to another system. After the first twenty-four trials, a scheme was drawn up, commencing with a series of trials of triple-expansion at 200 pounds boiler pressure per square inch, with and without steam in the jackets. This series, involving thirty-two trials, was commenced in October, 1888, and completed in April, 1889. Several trials were made at each speed and the results were found to agree within one per cent., so that three trials with steam in

the jackets and three without were taken as illustrating the results obtained. The conditions under which this series of trials had been made were, if anything, more favorable to economy than were any which prevailed in practice; and although the purpose of these engines was to elucidate the causes of inefficiency rather than to realize the utmost economy, yet it was very desirable that the results obtained should not, whether on account of the comparatively small sizes of the engines, or from other causes, fall greatly behind what might be expected from high-class engines in actual practice. Hence it was eminently satisfactory to find that, notwithstanding the drawbacks already mentioned, the economic results compared favorably with anything yet obtained in practice, even with the largest engines. The pounds of coal and of water per indicated horse power per hour were:

	With Steam Jackets.	Without Steam Jackets.
Coal—Total	1.50 to 1.33	1.81 to 1.62
Discounting radiation	1.30 to 1.21	1.77 to 1.54
Water—Total	14.10 to 12.68	17.30 to 15.90
Discounting radiation	12.30 to 11.90	16.60 to 15.10

Although these results were extremely good, the sources and extent of the various losses were clearly shown. Thus of the total heat received by the engines, exclusive of radiation, with jackets 19.4 per cent. had been converted into work, and without jackets 15.5 per cent.—the

greatest amount which would have been converted had there been no secondary actions being 23 per cent.; so that with steam jackets there were losses through secondary actions amounting to 17 per cent., and without jackets to 34 per cent. The manner of the distribution of these losses was also apparent. One important source of loss, which with jackets accounted for 5 per cent. of the loss, had been brought to light for the first time. This was the heat carried away from the surfaces of the cylinder and passages, in consequence of the expansion after release. The effects of cylinder condensation were clearly shown in the mean diagrams taken from the trials. Although these trials were not in themselves sufficient to determine anything like a complete law of this action, they exhibited in a striking manner its dependence on certain circumstances. One circumstance in particular, which had not previously received much attention, was here shown to be of primary importance in the action of steam jackets. These diagrams showed that with the temperature of steam in the jackets of No. I engine the same as that of the initial steam, the effect of the jackets on the cylinder condensation was very small. In No. II engine, with 80° Fah. difference in the temperature of the jackets and that of the initial steam, the condensation was reduced from 30 per cent. to 5 per cent.; and a difference of tem-

perature of 180° Fah. between the jackets and the initial steam in engine No. III entirely prevented condensation. Thus, in these trials, with steam at boiler pressure in the jackets, low-pressure diagrams had been obtained, apparently for the first time, in which the curve of expansion coincided exactly with the curve for saturated steam.

CHAPTER IV.

TRIPLE EXPANSION MARINE ENGINES.*

THE last few years may be regarded as a transition period in the history of marine engineering, as the high-pressure triple-expansion engine has now proved the successful rival of the double expansion compound. The object here is to bring forward the results of experience with this new type of engine, and to consider briefly the various points which have a direct bearing on its efficiency, as well as the most suitable design for marine purposes.

Position of Cylinders.—There has been great difference of opinion regarding the best method of placing the three cylinders in a triple expansion engine of ordinary size; and some very crude plans have been adopted since the introduction of the system by Mr. Kirk in the *Propontis.* The high pressure cylinder placed on the top of either the intermediate pressure or the low pressure has been tried, together with many schemes to lessen the difficulties of overhauling; while the main objections to this design appear to have been overlooked. With the exception

* Wyllie.

of what was being done in the matter by Mr. Kirk, no further attempt seems to have been made to construct small power expansion engines on three cranks until about three years ago, when the writer undertook to build one of 700 indicated horse power which should fulfill the condition that no more space was to be occupied than would have been taken up with an ordinary compound. Engines of small power had previously been constructed on the tandem principle; but experience had shown the writer that in order to take full advantage of the triple expansion system, an engine must be built on three cranks placed at equal angles; and the best proof that this was the correct solution of the problem is the fact that the arrangement is now almost universally followed, although, at the time, it was considered by the advocates of the tandem engine and others to be a step entirely in the wrong direction.

General Conditions of Efficiency.—The most important conditions to be considered in order to obtain an efficient engine are that there should be approximate equality, first, in the range of temperature in each cylinder; secondly, in the initial stress on each crank; and thirdly, in the indicated horse power of each engine. What may be termed the complements to these three essentials are: 1, steam jacketed cylinders; 2, cylinder ratios; 3, velocities of initial and ex-

haust steam; 4, clearance and compression; 5, receiver capacity; 6, piston speed; 7, order of sequence of cranks.

Steam Jackets.—The subject of steam jacketing, and fitting cylinders with working barrels cast separately, has undergone considerable discussion among engineers, although the theoretical advantages of the system have long since been admitted. Keen competition seems evidently to have been one of the reasons why this fitting in many instances has been discontinued, in order to meet the demand for a cheap engine. Thorough investigation of the subject is beyond our present scope; but the action of the cylinder surface, producing, as it does, an influence on coal consumption, deserves more attention than it often receives. In a single cylinder engine with a high ratio of expansion and a large range of temperature, it is well known that in each stroke there is excessive initial condensation, followed by partial re-evaporation. In triple expansion engines, where attention is paid to the equal division of the total range of temperature among the cylinders in which the successive stages of expansion take place, the benefits arising from the use of jackets are naturally not so great as in a single cylinder engine with a high ratio of expansion; but however carefully the triple engine may be designed, the jacketing of at least the intermediate and low pressure cylin-

ders is essential to maximum efficiency. Beyond the actual economy resulting from steam jacketing, there are numerous practical advantages of a working barrel cast separately. The cylinder barrel being one of the most vital parts of an engine, it is essential that it should be of the finest material and of uniform quality, in order to prevent it from wearing unevenly, as an oval or uneven cylinder means a leaky piston and increased coal consumption. In the case of a cracked cylinder without an independent liner, the expense of a renewal is very great, while a new liner costs but little; and the additional facility given for heating up cylinders fitted with independent liners is a safeguard agninst accidents of that description.

Cylinder Ratios.—The ratio of cylinder capacities in a triple expansion engine depends on the pressure of steam and the type of engine. In cargo steamers, where economy of fuel is of vital importance and a large range of reserve power is not necessary, the high-pressure cylinder should be of such a diameter that with a cut-off of from 50 to 60 per cent. the theoretical absolute terminal pressure in the low-pressure cylinder shall not exceed 10 lb. per square inch. In war ships, where a large range of power is sometimes required and economy of fuel is not so important, the high-pressure cylinder should be larger in proportion, so that a higher mean

pressure can be obtained. In triple expansion engines on three cranks, the intermediate cylinder should be so proportioned that, with 55 to 65 per cent. cut off, the powers, the ranges of temperature and the initial stresses in the three cylinders may approach equality.

Steam Velocities.—To obtain even approximate equality in powers, temperatures, and stresses, requires the greatest care in designing the steam passages throughout the engine; and unless the velocities of the steam at the various points and the degrees of cut-off by the valves are carefully proportioned, it will be found that these three elements of economy and of efficient working are very far from being realized. The greater number of published results show that the subject is still inadequately appreciated. An expanded high pressure diagram taken from an engine with 135 lb. boiler pressure shows that the velocity in the passages was by no means high, but the pipes were indirect, and owing to specified requirements there were several altogether unnecessary valves; the result was a fall of pressure to 120 lb., or a loss of 15 lb. available initial pressure. When the objectionable bends were removed, the initial pressure in the diagram rose to 130 lb., thus raising the ratio of expansion and increasing the general efficiency. The stroke of the engine was 3 ft., and the revolutions 104 per minute. A high pressure ex-

panded diagram from a cylinder of the same diameter and stroke shows the revolutions to be 70 per minute. This shows how small the initial drop may be, when the sources of loss are thoroughly appreciated.

Piston Valves.—The use of piston valves on the intermediate and low pressure cylinders, especially in engines of moderate power, serves to illustrate how the indirectness of the valve passages impairs the efficiency of the steam, and in some cases more than balances the beneficial effect of reducen friction in the machinery. An expanded diagram taken from the intermediate cylinder of a triple engine fitted with a piston valve shows considerable wire-drawing, which is to be attributed to the indirectness of the valve passages; for in another diagram, taken from a similar engine in every respect, with only a slightly lower speed of steam but with an ordinary slide, the wire drawing is much less.

Low Pressure Cylinders.—The low pressure cylinder is of course the source of the greatest inefficiency; and too much care cannot be taken in the design of the steam ports and exhaust passages. The steam passage should be as short as possible, so as to reduce the clearance to a minimum; and the speed of the entering steam should not be so high as to cause excessive frictional resistance, nor the speed of exhaust so high as to augment the back pressure; conse-

quently, the greatest efficiency is obtained when the revolutions and indicated horse power are not required to vary to any great extent. As an illustration, a diagram was taken from a low-pressure cylinder which was designed for 60 to 65 revolutions per minute; on the lightship trial the engines were run at 78 revolutions, and the vacuum was then 1¾ lb. less than with the same cut off at 68 revolutions, the loss being entirely due to the excessive exhaust velocity consequent on driving the engines at a greater speed than they were designed for. Contracted or indirect exhaust passages in the high pressure and intermediate cylinders have the effect of causing a larger difference between the back pressure on one piston and the initial pressure on the next, thus diminishing the efficiency of the steam. An illustration of this defect is furnished by the diagrams which were taken from a triple expansion engine built in the North of England.

Cut-Off.—The cut-off necessary to maintain the equality of the three essentials for an efficient engine is governed to a great extent by the speed of the entering steam and the nature of the passages, inasmuch as the same amount of steam per stroke may enter the cylinder at the ordinary working number of revolutions per minute, as at a higher speed with later cut-off and larger port opening; and in the latter case

the difference will be greater between the actual and the effective cut-off. An expanded high pressure diagram was taken on a lightship trial, and shows the effect produced by a high velocity of steam on the volume admitted at each stroke, the actual cut-off in this case being 8 per cent. later in the stroke than the virtual or effective cut-off. In the intermediate and low pressure cylinders, too high a velocity of the entering steam will produce excessive frictional resistance, causing a drop in the expansion curve as well as an unduly high receiver pressure, thus disturbing the equality of temperatures and of initial stresses. A diagram was taken from the intermediate cylinder on a lightship trial with a high steam velocity through the ports, and another diagram from the same cylinder in the loaded ship; the cut off was equal in both instances, yet in the latter case the receiver pressure is 7 lb. lower, and the drop in the expansion curve is very considerably reduced. A three crank engine, with its more uniform twisting moment on the shaft, and its approximately equal initial loads, presents every facility for increased revolutions or higher piston speed; and as lighter machinery for the same power developed admits of additional capacity for carrying cargo, the recognized standard of sixty revolutions per minute, so indelibly stamped on marine engineering practice, is unfavorable to the greatest

economy. It is true that the absurdly bluff run aft which is found in many steamers interferes with a high number of revolutions; but there is no reason why a considerable advance should not be made. Engineers are often severely handicapped and sometimes utterly baffled by the design of the cargo "receptacle" which has to be propelled; and should the speed not reach the unreasonable expectations formed, the general inefficiency of the engines is at once set down as the cause, although the ship may have a displacement co-efficient as high as 0.77 and may never run a straight course except by accident. That ships having a large cargo carrying capacity and traveling at a low speed are pecuniarily successful is not to be denied; but that beneficial results would follow from a slightly less box-like form is abundantly evident, though not yet universally appreciated in practice.

Sequence of Cranks.—In a triple engine some diversity of opinion seems to have existed as to the order of sequence for the three cranks. Many engines are arranged with the high pressure crank leading, followed by the intermediate crank, and the low pressure crank last; which causes excessive variation in the receiver pressure, increased ranges of temperature in the cylinders and considerable differences in initial stresses. The better sequence is high pressure leading, low pressure following, and intermedi-

ate last. Expanded diagrams were taken from an engine built on the Tyne with the common arrangement of cranks, namely, high, intermediate, and low; here the pressure in the receiver into which the high-pressure cylinder exhausts varies as much as 22½ lbs., this large amount being probably due to the receiver being too small. The temperatures and stresses in the high pressure and intermediate cylinders approach equality; but the amount in each case is much greater than it would have been had the low pressure crank followed the high pressure. Other diagrams taken from an engine built in Scotland show the same sequence of cranks, namely, high, intermediate and low; but the receiver from the high pressure cylinder being here larger, the variation of pressure in it is reduced to 12 lb. In this engine, however, the temperatures and stresses are astray, the effects of the latter being shown in the crank-shaft diagram. In the next diagram the low pressure cylinder follows the high pressure, with the evident beneficial effect of reducing the variation of pressure in the first receiver to 6 lb., and also reducing the ranges of temperature in the cylinders, as well as the initial loads on the pistons.

Number of Cranks.—The question of the most suitable type of marine engine resolves itself into two general divisions, namely, engines working on two cranks, and those working

on three. Considerable difference of opinion has existed as to which is the better, and many engineers have decided in favor of the two-crank tandem, the high pressure cylinder being placed over either the intermediate or the low pressure. A preferable arrangement is to place two cylinders on each crank. But both these plans present very objectionable features. In a three-cylinder tandem it is impossible to obtain anything approaching equality of temperatures, stresses, and powers; and therefore the result is a loss of efficiency. In converting compound into triple-expansion engines, the method of adding another cylinder on the top of the high pressure or low pressure is certainly an easy way of applying the triple expansion principle, but for obvious reasons it is a very objectionable one. In the diagrams from a tandem engine, in which every possible care was taken in the design, the results do not compare at all favorably with those from a good three-crank engine. To have two cylinders on each crank is, undoubtedly, the best design for a two-crank engine on the triple expansion principle, as it is then possible to get an approximately equal initial stress on each crank. This arrangement, of course, necessitates one of the three stages of expansion taking place in two cylinders, instead of in one. The diagrams from this type of engine show the range of temperature, varying from 59° to 81°,

while the crank-shaft diagram is even worse than that from an ordinary compound on two cranks. A marine engine should be so designed that any working part can be easily examined or removed, and this is impossible with a tandem engine.

The arrangement of cylinders on three cranks fulfills the required conditions more nearly than any other design; and that this is appreciated by many engineers and ship owners is evident from the rapidity with which the three-crank engines are entirely displacing the two-crank. A few of the principal advantages are: (1) More uniform strain on shafting; (2) adaptability for a higher rate of revolution and increased piston speed, thus obtaining increased efficiency from the steam in the engine, as well as lighter machinery in proportion to the power developed; (3) less wear and tear; (4) easier accessibility of working parts; (5) interchangeability of parts, thus minimizing the consequences of a breakdown; (6) greater facility for repairs; (7) easier adjustment of temperatures, stresses, and powers. All these features are deserving of investigation.

Regarding the more uniform twisting moment on the crank-shaft of a three-crank engine, it is often advanced that the difference between crank-shaft diagrams from a two-crank and a three-crank engine is not so great as to cause

any appreciable effect. It must be remembered, however, that the variable twisting strains on the crank-shaft are reflected to a greater or less extent throughout all the working parts of the engine; and it seems certain that the beneficial influence of three cranks is far beyond what is generally accepted. There seems to be an impression that the length of an engine-room is increased by the introduction of three cranks. But by placing the valve casings at the side, thus allowing the intermediate and low-pressure cylinder covers to be brought close together, the total length of a three-crank engine is made no greater than in nine-tenths of the existing compounds. In the Orient liner *Lusitania*, since she has been converted to three cranks by the writer, the distance longitudinally over the cylinders is one foot shorter than before, and the indicated horse power is 800 greater.

Valve Gear.—The arrangements for working valves are becoming very extensive, and a review of the various devices which are employed to distribute the steam in modern engines would by itself occupy a very lengthy paper. The requirements of a good valve gear are that it shall give at both ends of the cylinder an equal distribution of steam at all grades of expansion, with a minimum of working parts and with no undue strains. The four principal methods adopted to work valves are: first, by the single

eccentric; secondly, by the double eccentric; thirdly, by taking the motion from the connecting rod; and fourthly, by a compound motion derived from both the piston rod and the connecting rod. All four of these have their advantages and defects, and vary considerably in complexity and in multiplicity of parts. The single eccentric valve gear, giving almost perfect steam distribution, having few working parts, and being independent of the connecting rod or piston rod, seems eminently suitable to fulfill all the desired conditions. A sliding block gear of this kind has given every satisfaction in actual working; and the arrangement is such as to allow free access to all the working parts, while occupying a minimum of space. The end of the eccentric rod reciprocates on guide bars, which are inclined at various angles to suit the desired action of the valve. Motion is imparted to the valve spindle from an intermediate joint in the eccentric rod, which moves in an approximately elliptic path. A slot in the tail of the guide bars admits of varying their inclination to different extents for a given movement of the reversing engine, thus altering independently the cut-off in each cylinder. The long leverage and easy motion reduce the wear and tear to a minimum, as results in actual practice have proved. An objectionable feature, however, in an engine fitted with this sliding block gear is that the

valves are at the front, over the starting platform; and the exhaust has to be led by a belt around the low pressure cylinder to the condenser. To overcome this objection of the sliding block gear, a swinging link gear was designed. The eccentric rod, as in the last case, is placed diagonally over the condenser, but is here guided in an arc of a circle, by suspending it by a swinging link, centered on a pin, which pin is adjustable by the reversing engine into various positions for varying the grade of expansion either ahead or astern.

The movements for working the valve are transmitted from a joint at the end of the eccentric rod by a compensating link connecting the joint with one arm of an oblique lever, of which the other arm is jointed to the valve spindle. The compensating link is an essential and distinguishing feature of this gear; it is so placed and proportioned relatively to the other parts as to produce practically equal port opening and cut-off at each end of the stroke. There is a quick and a slow movement of the valve at each end of its travel; the slow movement being at the maximum port opening, and the quick movement at the cut-off. The lead is also constant at all grades of expansion.

Practical Results.—In the engines of the screw-steamer *Para*, belonging to Messrs. Steel, Young & Co., which made her maiden voyage

to the River Plata about 3½ years ago, the cylinders are 19 inches, 35 inches, and 53 inches in diameter, with 33-inch stroke. These being the writer's first triple-expansion engines, special arrangements were made for ascertaining the actual coal consumption per indicated horsepower. Although, in order to get a correct estimate of an engine's performance, the quantity and temperature of the feed and circulating water ought of course to be considered, yet there are so many practical difficulties in the way of getting these particulars on board ship, that it has been found impossible to obtain any reliable data on these points. This steamer still continues on the same run, averaging 9 knots an hour on 10¼ tons of coal; and has not yet cost anything beyond the usual overhaul for repairs. The great saving in coal consumption with the triple engines is apparent when comparison is made with two sister ships fitted with compound engines, the *Ingram* and the *Wandle*, belonging to the same company, and built by the same builders; the comparison is shown in Table I., which gives the average working over a period of three years.

TABLE I.

Comparative Results from Three Similar Steamers with Compound and with Triple Expansion Engines.

Name of screw steamer	Ingram.	Wandle.	Para.
Length, feet and inches	257 6	257 6	257 6
Breadth, feet and inches	34 6	34 6	34 6
Draught, feet and inches	18 10½	18 6½	19 4
Dead weight carried, tons	2,310	2,203	2,398
Type of engines	Compound.	Compound.	Triple.
Boiler pressure, lb. per square inch	75	75	150
Speed, knots per hour	8½	8½	9
Indicated horse power, total	570	580	620
Coal consumption per day, tons	13¾	14	10¼

The diagrams are an expanded set taken during the three days' trial of the *Para;* but owing to the intermediate cylinder being rather too large, the equalities of temperatures and of initial stresses are disturbed; and the drop of the steam pressure in the intermediate cylinder is excessive, in consequence of the steam velocity being too great.

In Table II. is given a comparative statement of results from an approximately similar trio of boats in the same trade and under the same management, on a round voyage to Java under average conditions. The triple engines in the *Jacatra* are of the same general design as those in the *Para*, but of greater power; the compounds in the two other boats are of the ordinary type.

TABLE II.

Comparative Results from Three Steamers with Compound and with Triple Expansion Engines.

Name of screw steamer	Fellinger.	Padang.	Jacatra
Length, feet and inches	286 2	300 0	314 0
Breadth, feet and inches	35 0	37 0	37 9
Draught, feet and inches	20 3	21 9	21 6
Dead weight carried, tons	2,600	3,000	3,300
Type of engines	Compound.	Compound.	Triple.
Boiler pressure. lb. per square inch	70	76	140
Speed, knots per hour	9	9½	10
Indicated horse power, total	600	790	890
Coal consumption per day, tons	15½	18	13½
Ditto per I. H. P. per hour. lb.	2.19	2.13	1.41
Quality of coal used	German.	Cardiff.	Mixed.

The design of the triple engines of the Union Company's screw steamer *African* and Messrs. Glover Brothers' screw steamer *Shakespear* differs from that in the *Para* and *Jacatra* in having the valves situated over the condenser, thus doing away with the exhaust belt around the low pressure cylinder, and giving a free, open front to the engines; all the working parts are thereby rendered easily accessible, with every facility for overhauling. The expanded diagrams taken from the *Shakespear* under average working conditions have already been referred to. The equalities of temperatures, initial stresses, and powers are as near as it is practically possible to get them; and this result has been obtained by carefully considering the various points which influence the efficiency of the engine. The expanded diagrams taken on the trial trip of the *African* in Stokes Bay show 1,086 horse power, developed at 85½ revolutions

per minute. They thus prove that there is a sacrifice of weight in triple engines when making no more than 60 revolutions per minute, inasmuch as the power developed at 66 revolutions per minute by the duplicate engines of the *Shakespear* is much less, namely, 871 horse power. The expanded diagrams from the original compound engines of the Union Company's screw steamer *Anglian* were taken under average conditions on a voyage to the Cape. The mean consumption of coal per day over eight voyages was 24 tons, or about 2.1 lb. per indicated horse power per hour. The compound engines were of the ordinary two-cylinder type, the valves being directly over the screw-shaft, and driven by the usual link motion. Diagrams were taken on a trial trip after the engines had been converted to triple expansion; and on her voyage to the Cape, the average speed being exactly the same as in the eight voyages above referred to, the coal consumption was one-third less, namely, 16 tons; and as she is placed on a foreign station where the cost of fuel is about $10 per ton, the economy of the conversion is obvious. The method of converting consisted in replacing the old high pressure cylinder by a new intermediate cylinder, and adding a new high pressure engine complete on the forward end of the screw-shaft, the high pressure and intermediate valves being driven by sliding-block gear. By

arranging the valve casings at the side of the high pressure and intermediate cylinders, the distance fore and aft over the present engines is very little more than before, although the power is now sufficient to drive the vessel one knot an hour faster than her former maximum speed. The original compound engines of the Orient Company's screw steamer *Lusitania* were of the two-cylinder type, with an expansion valve on the high pressure cylinder. The diameters of the cylinders were 58 and 103 inches, with a stroke of 4 feet; and the boiler pressure was 55 lb. per square inch. The diagrams were taken under ordinary working conditions on a voyage from London to Sydney, the average daily consumption being 52 tons of Welsh coal. The old cylinders were afterward replaced by new ones, and a high pressure engine was added complete, its valve being worked by swinging link gear. By arranging the intermediate and low pressure slide valves between the intermediate and low pressure cylinders, the old gear was utilized; and by reducing the valve casings to modern proportions, the distance lengthwise over all the three new cylinders is one foot less than before. Subsequent diagrams represent the present average working conditions of the engines, the increased power propelling the vessel at a much higher speed. The coal consumption at this power is about 50 tons per day; but traveling at

the former speed, the consumption is reduced from 52 tons to 37 tons; so that, allowing eighty steaming days for a return voyage to Australia, the saving of coal is 1,200 tons, while the cargo-carrying capacity is largely increased. The foregoing results represent a fair average of those obtained from thirty triple expansion engines, all of the three-crank type, which have been designed by the writer during the last three years; and are sufficient to prove that this kind of engine is most efficient, and is undoubtedly the engine of the future.

Artificial Draught for Boilers.—The next step in marine engineering is probably the application of artificial draught to boiler furnaces. Expanded diagrams were taken from the engines of the screw steamer *Stella*, belonging to Messrs. Herskind & Woods, between Hartlepool and Dover, on her voyage to Bombay; and as a steady boiler-pressure is always maintained, the engine power may be considered as constant. The average speed between port and port from Hartlepool to Bombay was fully 9 knots per hour, the dead weight carried was 3,680 tons, and the daily consumption of North Country coal was 13.6 tons. This performance is very remarkable, when the dimensions of the ship are considered, namely, length 302 feet, breadth 38 feet, and displacement coefficient as high as 0.77. The special features of the arrangement

are the method employed for heating the air both outside and inside the uptake, and the application of balanced firedoors, which on being opened shut off automatically the hot air supplied by the fan, both above and below the fire bars. This is the first application of artificial draught to the boilers of triple expansion engines; and should the results fulfill expectations, there is little doubt that the plan will be extensively adopted, as being yet another step toward economy of fuel.

CHAPTER V.

CYLINDER RATIOS OF TRIPLE EXPANSION ENGINES.*

WHENEVER the steam pressure exceeds 100 lbs. gauge, the triple expansion engine is used, while the tendency is toward quadruple expansion when the pressure exceeds 170 lbs.

Existing practice in proportioning the cylinders of triple expansion engines is given in Table I (page 89). In it the particulars of eighty engines of recent design are given, the engines being grouped according to boiler pressures. The tendency, as shown, is toward an increase in piston speed and boiler pressure, and a consequent decrease in weight and first cost of machinery. Equal work should be performed in each cylinder, uniform rotative effort secured, and low initial strains in all moving parts obtained. These can be nearly obtained by dividing the work among three or more cylinders, as in the triple expansion engine, and by the use of variable expansion valves and balanced rotative parts.

When the piston speed varies from 750 to 1,000 feet per minute, the following cylinder

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* Whitman.

ratios are recommended as the result of a study of Table I. (the terminal pressure of steam in the large cylinder being about 10 lbs. absolute), viz:

CYLINDER RATIOS RECOMMENDED FOR TRIPLE-EXPANSION ENGINES.

Boiler Pressure (Gauge.)	CYLINDER RATIOS.		
	Small.	Intermediate.	Large.
130	1	2.25	5.00
140	1	2.40	5.85
150	1	2.55	6.90
160	1	2.70	7.25
170 and upwards—quadruple-expansion engine to be used.			

Two methods of ascertaining the diameter of pistons will now be given.

1. *Annular Ring Method.*—In Fig. 3 lay down a theoretical indicator diagram of a simple engine for the particular expansion desired. Lay off the back-pressure line as shown. By trial find (with the polar planimeter or otherwise) the position of the lines *D E* and *F G* such that the areas marked "*A*" "*B*" and "*C*," are respectively equal. Find the mean ordinate of each area: that of "*C*" will be the mean unbalanced pressure on the small piston; that of "*B*" will be the mean unbalanced pressure on the area remaining after subtracting the area of the small piston from that of the intermediate; and that of the area "*A*" will denote the mean unbalanced pressure on a square inch of the annual

ring of the large piston obtained by subtracting the intermediate from the large piston. We thus see that the mean ordinates of the two lower cards act on annular rings.

FIG. 3.

Let H = area of H. P. Piston in square inches.

Let I = area of Intermediate Piston in square inches,

Let L = area of L. P. Piston in square inches.

Let P_h = mean unbalanced pressure per square inch from card "*C.*"

Let P_i = mean unbalanced pressure per square inch from card "*B.*"

Let P_l = mean unbalanced pressure from card "*A.*"

Let S = piston speed in feet per minute.

$(I.H.P.)$ = indicated horse-power of engine. Then for equal work in each cylinder we have:

$$\text{Area of small piston} = H = \frac{33000 \times \frac{(I.H.P.)}{3}}{p_h \times S}. \quad (1.)$$

$$\left.\begin{array}{l}\text{Area of annular ring of}\\ \quad\text{intermediate cylinder}\end{array}\right\} = \frac{33000 \times \frac{(I.H.P.)}{3}}{p_i \times S}$$

Area of intermediate piston

$$= I = H + \frac{33000 \times \frac{(I.H.P.)}{3}}{p_i \times S}. \quad (2.)$$

Area of annular ring of large piston

$$= \frac{33000 \times \frac{(I.H.P.)}{3}}{p_l \times S}$$

Area of large piston

$$= L = I + \frac{33000 \times \frac{(I.H.P.)}{3}}{p_l \times S}. \quad (3.)$$

This method is illustrated by the following:

Example: Given $I.\ H.\ P. = 3000$, piston speed $S = 900$ feet per minute, ratio of expansion 10, initial steam pressure at cylinder 127 lbs. absolute, and back pressure in large cylinder 4 lbs. absolute. Find cylinder diameters for equal work in each (Fig. 3).

The mean ordinate of "*C*" is found to be $P_h = 37.414$ per square inch.

The mean ordinate of "*B*" is found to be $P_i = 15.782$ lbs. per square inch.

The mean ordinate of "*A*" is found to be P_l = 11.730 lbs. per square inch.

Then by (1), (2), and (3), we have—

$$H = \frac{33000 \times \frac{3000}{3}}{37.414 \times 900} = 980 \text{ sq. in., diameter } 35\tfrac{3}{8}''.$$

$$I = 980 + \frac{33000 \times \frac{3000}{3}}{15.782 \times 900} = 3303 \text{ sq. in., diameter } 65''.$$

$$L = 3303 + \frac{33000 \times \frac{3000}{3}}{11.730 \times 900} = 6432 \text{ sq. in., diameter } 90\tfrac{1}{2}''.$$

2. "*Drop*" *Method.*—In Fig. 4 lay down a theoretical card as in Fig. 3. Choose cylinder ratios from the table of recommended values. (Page 84.)

Draw *FG*, *DE*, and *JK* in these ratios, dividing the diagram in three parts, "*A*," "*B*," and "*C*." Round off the corners so as to make the figure conform as nearly as possible to the combined card from an engine of this type. The waste spaces are due to "drop," condensation, etc. The mean ordinate of this combined card will give the mean unbalanced pressure on a square inch of the large piston, as if all work had been done in its cylinder. With this find the area of the piston necessary for all the work. Divide this area of piston by the cylinder ratio, thus obtaining the area of each piston, as follows:

Example: Given data of previous example,

and cylinder ratios of 1 to 2.25 to 5.00, find diameter of each cylinder (Fig. 4.)

FIG. 4.

The mean ordinate of the combined card is measured to be 17.5 lbs., hence

Area of large piston

$$= L = \frac{3000 \times 33000}{17.5 \times 900} = 6286 \text{ sq. in., diameter } 89\tfrac{1}{2}''.$$

Area of intermediate piston

$$= I = \frac{2.25\, L}{5.00} = 2829 \text{ sq. in., diameter } 60''.$$

Area of small piston

$= H = \dfrac{L}{5.00} = 1257$ sq. in., diameter 40″.

The results obtained by the two methods accord quite closely. The main point to be considered is: How large shall the *L. P.* cylinder be? The *H. P.* cyl., and *I. P.* cyl. may have almost any values provided variable expansion valves are used.

CYLINDER RATIOS FOR TRIPLE EXPANSION ENGINES.

TABLE I.

Showing Cylinder Ratios of Triple-Expansion Engines for Variations in the Boiler Pressure.

VESSEL.	Cylinder Ratios.			Boiler Pressure Gauge	Piston Speed in Feet per Min.	I. H. P.
	H. P.	I. P.	L. P.			
Gunboat No. 1 . . .	1	2.00	5.17	160	. . .	3000
Montebello	1	2.40	5.70	160	1060	4200
Iljin	1	2.38	5.76	160	. . .	3550
Oroya.	1	2.72	6.25	160	774	6750
Orizaba	1	2.72	6.25	160	756	6500
Buffalo	1	2.68	6.80	160	570	2276
Bleville	1	2.47	6.13	160	490	1223
Vespasian	1	2.91	7.32	160	400	600
Aberdeen	1	2.81	9.00	160		
Altmore	1	2.68	9.00	160		
Anchoria	1	2.69	7.11	160		
Cosmopolitan . . .	1	2.69	7.11	160		
Cremon	1	2.48	6.25	160		
Earnholm	1	2.56	7.97	160		
Explora	1	2.72	7.54	160		
Gulf of Suez	1	2.69	7.11	160		
Hecla	1	2.71	7.64	160		
Mexican	1	2.60	6.82	160		
Pacifique	1	2.61	7.04	160		
Trojan	1	2.52	6.85	160		
Sevona	1	2.78	7.63	160		
Athenian	1	2.61	6.82	160		
Birkhall	1	3.32	11.20	160		
City of Lincoln . .	1	2.60	6.82	160		
De Ruter	1	3.27	9.87	160		
Etna	1	2.71	7.32	160		
Spartan	1	2.52	6.85	160		
Average cyl. ratios for 160 lbs. boiler pres.	1	2.66	7.24			

CYLINDER RATIOS FOR TRIPLE EXPANSION ENGINES.—*Continued.*

Vessel.	Piston Stroke in Inches.	Cylinder Diameter in Inches.			Cylinder Ratios.			Boiler Pressure Gauge	Piston Speed in Feet per Min.	I. H. P.
		H. P.	I. P.	L. P.	H. P.	I. P.	L. P.			
Dogali	33	--	-	73	1	2.25	5 92	150	842	7600
Ormus	72			112	1	2.52	5 93	150	840	8000
Aller	72			108	1	2 53	6 03	150	828	6890
Saale	72			108	1	2.53	6.03	150	828	6890
Trave	72			108	1	2.53	6 03	150	828	6890
Trans-Pacific	60			90	1	2.71	7 01	150	. .	4000
Carmarthenshire	45			70	1	2.54	6.72	150	568	2340
Libra	42			67	1	2.82	7 18	150	616	1966
Sobralense	42			60	1	2.50	6.23	150	532	1580
Westmoreland	36			54	1	2.72	7 29	150	420	900
Royal Prince	39			54	1	2.72	7.29	150	481	767
City of Edinburgh	48			73	1	2.15	5.92	150		
Congella	42			56	1	2.62	7.11	150		
Constantin	. .			34	1	2.69	7 40	150		
Ehrenfels	48			72	1	2.31	8 29	150		
Florence Richards	30			48	1	2.13	6 73	150		
Thames	36			54	1	2.26	6 31	150		
Vlissingen	18			26	1	2.56	6 76	150		
Asturiano	36			55	1	2.51	6 54	150		
Drummond Castle	57			88	1	4.09	7 11	150		
Elbe	48			88	1	2.58	7.11	150		
Ennerdale	33			52	1	2.56	9.36	150		
Grantully Castle	57			82	1	2.71	7.00	150		
Jenny Otto	36			57	1	2.56	8.12	150		
Ocean King	48			68	1	2.61	6.84	150		
Russia	45			70	1	2.34	7 25	150		
Remo	33			47	1	2.75	7 21	150		
Yeoman	44			63	1	2.31	6.35	150		
Average cyl. ratios for 150 *lbs. boiler pres.*					1	2.54	6.90			
Destructor	21	18½	27	42	1	2.13	5.16	145	1023	3829
Oaxaca	60	40	64	92	1	2.56	5.29	145	700	3417
Average cyl. ratios for 145 *lbs. boiler pres.*					1	2.35	5.23			
Serpent	33		37	57	1	2.03	4 81	140	. . .	4500
Racoon	33		37	57	1	2.03	4 81	140	. . .	4500
Bramble	24		29	43	1	2.51	5 40	140	732	1042
Lizard	24		29	43	1	2.51	5.40	140	732	1025
Rattler	24		29	43	1	2.51	5.40	140	810	1291
Wasp	24		29	43	1	2.51	5 40	140	772	1157
Rattlesnake	18		27	42	1	2.13	5 16	140	966	2860
Reina Regente	45		60	92	1	2 25	5.28	140	862	12000
Windsor	33		30	56	1	2.75	9.68	140	. . .	
Bengal	66		56	89	1	2.56	6.47	140	. . .	3200
Coromandel	66		56	89	1	2.56	6.47	140	. . .	3200
Average cyl. ratios for 140 *lbs. boiler pres.*					1	2.40	5 84			

CYLINDER RATIOS FOR TRIPLE EXPANSION ENGINES.—*Continued.*

Average cyl. ratio for 135 lbs. boiler pres.					1	2.07	5.00			
Victoria	48	38	58	88	1	2.33	5.36	130	. . .	12000
Sanspareil	48	38	58	88	1	2.33	5.36	130	. . .	12000
Aurora	42	36	51	78	1	2.00	4.70	130	. . .	8500
Australia	42	36	51	78	1	2.00	4.70	130	. . .	8500
Galatea	42	36	51	78	1	2.00	4.70	130	. . .	8500
Immortalite	42	36	51	78	1	2.00	4.70	130	. . .	8500
Orlando	42	36	52	78	1	2.08	4.70	130	832	8662
Undaunted	42	36	52	78	1	2.08	4.70	130	. . .	8500
Narcissus	42	35	51	78	1	2.12	4.97	130	. . .	8500
Average cyl. ratios for 130 lbs. boiler pres.					1	2.10	4.88			

CHAPTER VI.

CALCULATION OF WORK DONE IN A COMPOUND ENGINE.*

IN the working of a compound engine, where the small cylinder exhausts into the large one, the work done in a stroke depends on the size of the large cylinder, and is the same as that which would be performed in a single cylinder of the same content, by expanding to the same extent from a like initial pressure.

This proposition is easily proved, and any one may satisfy himself that it is approximately true by examining a well-formed indicator diagram as taken from a compound engine.

Referring to Fig. 5, which is from an engine having a high-pressure cylinder 18 inches in diameter, with a 6-feet stroke, and a low-pressure cylinder of 36 inches in diameter, with a stroke also of 6 feet, the number of revolutions being 34 per minute.

Since the lengths of stroke are the same, and the areas of the pistons are as 1 to 4, it follows that the indicator diagram marked *A*, as taken from the high-pressure cylinder, would be reduced to the same scale as that from the low-

* Goodeve.

pressure cylinder marked B, if we supposed the diameter of the latter cylinder to be 36 inches and the stroke 5 feet, or one-fourth of that which it really is.

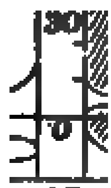

FIG. 5.

This result is set out in Fig. 6

The diagram marked A is reversed in position and repeated on the right-hand side by measuring off a series of horizontal lines, such as $c\ d$, and making $c\ d$ equal $\frac{1}{4}$ $C\ D$ in every case.

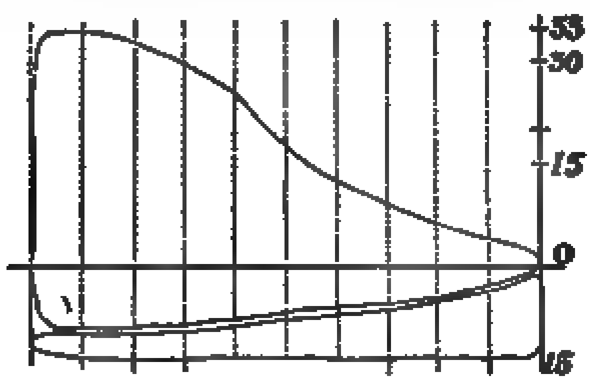

FIG. 6.

In this way the upper shaded area represents the work done in the high-pressure cylinder as

it would appear on the scale adopted in the low-pressure cylinder. The bottom shaded area is merely a repetition of the area *B*.

When the two diagrams are put together, it will be seen that the two portions of the expansion curves fit very fairly or run into one, and that the expansion commenced above *d* is carried on throughout the stroke.

It will be noticed that there is a little want of similarity between this diagram and Fig. 7. Here the steam line in *b* is horizontal at first, and then slopes downwards.

That it is horizontal at all is owing to some peculiarity in passing the steam from one cylinder to the other, as there should be the slope of an expansion curve throughout. But any deviation from theoretical proportions does not affect the general inference to be drawn from the two diagrams when viewed together, and we see that the expansion which has occurred in the high-pressure cylinder might very well have taken place in the low-pressure cylinder, as something which preceded the actual expansion therein.

ENGINES WITH CRANKS AT RIGHT ANGLES.

For many purposes it is enough to have an engine with a single steam cylinder, or an equivalent engine, with a pair of cylinders acting as one only; but on the other hand, there are numerous instances where two engines should be

placed side by side and work cranks at right angles to each other.

This is particularly the case in applying steam-power where it is of consequence to preserve the rotative pressure on the crank as nearly uniform as possible, and to maintain a smooth and even motion; or again in marine engines, for convenience of starting in any position, the same rule would hold; and before proceeding further it may be useful to point out the reason for the greater uniformity of rotative pressure which is a consequence of working with a pair of cranks at right angles.

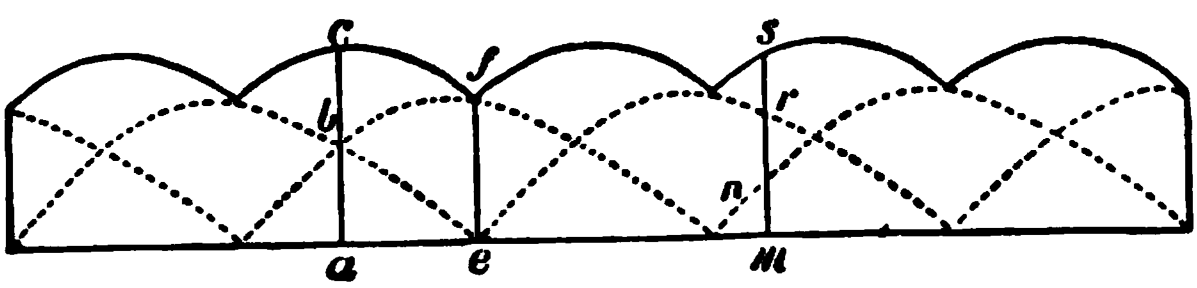

FIG. 7.

That the variations of tangential pressure on the crank of a direct-acting engine are represented by the vertical lines on a diagram similar to that shown by the dotted curve *b f* in Fig. 7.

Putting a series of such curves end to end, we obtain a graphical indication of the fluctuations of tangential pressure during the working of an engine with one cylinder.

The force is zero at dead point, and rises to *f e*, its greatest value, after which it sinks again to zero.

But if there be a pair of cranks at right angles, a second series of diagrams of rotative pressures must be superposed upon the first series, as shown by the second set of dotted curves, whereof one portion is marked *b e*, and the final result is exhibited by the upper line, not dotted, which is obtained by adding together the pairs of ordinates at each point. For example:

N M+M R=M S.
A B+A B=A C.
E F+O=E F.

The greater uniformity of rotative force is apparent, and it would be improved by cutting off at half stroke in each cylinder, for then the curve *b e* would be hollowed out and reduced, while the part *b f* would be unaffected, and the upper resultant wavy line would become more nearly horizontal.

By proceeding in this manner it is easy to set out a diagram of the rotative pressure upon the cranks of any pair of engines working under given conditions.

In applying these principles to direct-acting engines, where two cranks at right angles are to be connected with the cylinders, there are different methods for adoption, each of which has its advocates. One plan very commonly met with has been to place the high and low pressure cylinders in pairs, with their axes in the same straight line, so that one piston rod serves for

both. Thus, in marine engines, with the cylinders vertical, there may be:—

(1) The high-pressure cylinder above the low-pressure cylinder.

(2) The low-pressure cylinder at the top.

(3) The low-pressure cylinder encasing the high-pressure cylinder.

But in each of these cases, as also in compound horizontal engines, it is usual to confine the expansion to one pair of cylinders.

THE USE OF AN INTERMEDIATE RECEIVER.

In another class of compound engines there are two cranks at right angles, but only one cylinder connected with each crank.

Here each cylinder forms, as it were, an engine complete in itself; the cylinders (called A and B, as before) are placed side by side, and are of equal length; and the point to be noticed is, that the pistons in A and B no longer move together, but that one leads the other by half a stroke.

It is clear that the mode of exhausting at once from A into B is no longer applicable, and that some special method of distributing the steam, different from anything that we have yet seen, must be arranged.

The difficulty arises from the fact that the directions of motion of the pistons cross each other, whereby, for example, when the piston in

A is at the end of its stroke and about to ascend, that in B is in its middle position and is descending.

In order to get over this obstacle, Mr. Cowper has proposed to place an intermediate receiver between the cylinders *A* and *B*, which shall act as an exhaust reservoir for the steam coming from *A*, and as a boiler for the steam going into *B*. It appears that engines with a receiver have worked well in practice, but it seems difficult to justify the use of this arrangement by a strict reference to the principle of the theory of heat.

A general idea of the arrangement of the engine proposed by Mr. Cowper may be gathered

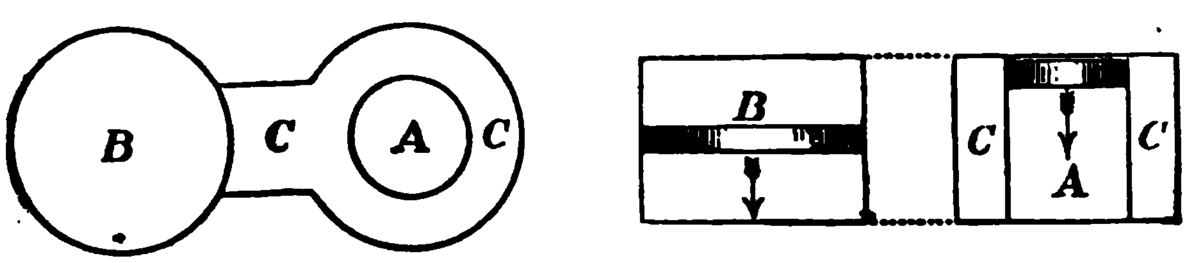

FIG. 8.

from the Fig. 8, where the cylinders *A* and *B* are placed side by side, and the high-pressure cylinder *A* is enveloped in a steam receiver or reservoir, marked *C*, the content of which is perhaps three times that of *A*.

In a working engine on this plan, steam (say at 70 lbs pressure) would enter *A* and be cut off at half-stroke; it would thus expand and finally exhaust itself into the receiver, where the pressure would vary from, say, 10 lbs. to 14 lbs.

The receiver would supply steam for the low-pressure cylinder *B*, just as if it were the boiler of an ordinary engine, and the pressure of the steam in *C* would fall to 10 lbs. when the demand upon it was made, but would rise to 14 lbs. when fresh steam entered it from *A*. The temperature of the steam in the jacket surrounding *A* is therefore much below that of the entering steam, which is so far a departure from the old practice.

CHAPTER VII.

TO FIND THE HORSE POWER OF SIMPLE, COMPOUND AND TRIPLE EXPANSION ENGINES.

It is customary to assume the expansion curve to be a rectangular hyperbola; and for the first approximation we calculate the horse-power of a compound or triple expansion engine of any sort as if the total expansion occurred in the cylinder or cylinders that exhaust into the condenser. Consequently, it is the large or low-pressure cylinder only which enters into the calculations. The initial steam pressure in the large cylinder we assume to be the same as that in the small cylinder or the cylinder into which the steam from the boiler first enters, and then expanded to the extent at which it enters the condenser. Here, then, we must know the rate of expansion, and when that is known, we must find the mean effective pressure. After this has been found, we apply the well-known rule for determining the *I. H. P.* of any simple engine, namely: $\frac{PLAN;}{33{,}000}$ in which P = mean effective steam pressure in pounds per square inch; L = length of stroke in feet;

A = area of piston in square inches; and N the number of strokes per minute, which is equal to the number of revolutions per minute multiplied by 2. Since, to find the mean effective pressure may be or appear to be the most difficult part of the calculation, it will be best to give an example in which the mean effective pressure is found when the initial steam pressure and the number of expansions are known, without any reference to the size of cylinders. Example: Let the initial steam pressure be 80 pounds, or $80 + 14.7 = 94.7$ absolute pressure, and let the pressure of the steam at the time it is exhausted into the condenser be 15.78 pounds absolute; back pressure 4 pounds. What will be the mean effective pressure per square inch, neglecting the effect of clearance and compression? Our first step will be to find the number of expansions, thus: Divide the absolute initial pressure by the absolute pressure at which the steam is exhausted; the quotient will be the number of expansions, hence, $\frac{94.7}{15.78} = 6$ expansions. To find the mean effective pressure, proceed as follows: Look for the number of expansions (6) in the table of hyperbolic logarithms + 1 and opposite it will be found 2.792; multiply this number by the absolute initial pressure, which will give a product of $2.792 \times 94.7 = 264.4024$; divide this product by the number of

expansions; the quotient will be the *mean pressure* per square inch; $\frac{264.4024}{6} = 44.06 +$ pounds. Subtracting the back pressure, we have a *mean effective pressure* of 44.06 pounds. In a similar way we may find the mean effective pressure of any compound engine.

Example 2. The large or low-pressure cylinder of a compound engine is 30 inches; stroke, 4 feet; number of revolutions per minute, 60; the mean effective pressure has been found to be 40.06 pounds per square inch; find the indicated horse-power. Applying the rule $\frac{PLAN}{33{,}000} =$ I. H. P., and remembering that the area of a piston 30 inches diameter is 706.86 square inches, we have

$$\frac{40.06 \times 4 \times 706.86 \times 60 \times 2}{33{,}000} = 412 \text{ indicated horse-power.}$$

A great deal of calculating can be dispensed with by using Table on page 105 to find the mean effective pressure, and table in Appendix to find area of piston or cylinder.

CHAPTER VIII.

TO FIND THE MEAN PRESSURE.

DIVIDE the length of the stroke by the length of the space into which the steam is admitted; find in the table the logarithm of the number nearest to the quotient, to which add 1—the sum is the ratio of the gain; then find the terminal pressure by dividing the initial pressure by the proportion of the stroke during which the steam is admitted, and multiply it by the logarithms + 1, found as above; the product will be the mean pressure through the stroke.

No.	Logarithm.	No.	Logarithm.	No.	Logarithm.
1.25	.22314	5.	1.60943	9.5	2.25129
1.5	.40546	5.25	1.65822	10.	2.30258
1.75	.55961	5.5	1.70474	11.	2.39789
2.	.69314	5.75	1.74919	12.	2.48490
2.25	.81093	6.	1.79175	13.	2.56494
2.5	.91629	6.25	1.83258	14.	2.63905
2.75	1.01160	6.5	1.87180	15.	2.76805
3.	1.09861	6.75	1.90954	16.	2.77258
3.25	1.17865	7.	1.94591	17.	2.83321
3.5	1.25276	7.25	1.98100	18.	2.89037
3.75	1.32175	7.5	2.01490	19.	2.94443
4.	1.38629	7.75	2.04769	20.	2.99573
4.25	1.44691	8.	2.07944	21.	3.04452
4.5	1.50507	8.5	2.14006	22.	3.09104
4.75	1.55814	9.	2.19722		

Example 1. Suppose the length of the stroke to be 48 inches, the initial pressure to be 40 pounds per square inch, and the steam to be cut off at 12 inches of the stroke, what will be the mean pressure?

$48 \div 12 = 4$. Hyp. log. of $4, = 1.38629 + 1 = 2.38629$. Then $40 \div 4 = 10 \times 2.38629 = 23.8629$ pounds, the mean pressure required.

Example 2. Suppose the length of the stroke to be 36″, initial pressure to be 50 pounds per square inch, and the steam to be cut off at 9″ of the stroke, what will be the average pressure?

$36 \div 9 = 4$. Hyp. log. of $4 = 1.38629 + 1 = 2.38629$. Then $50 \div 4 = 12.5 \times 2.38 = 29.75$, mean pressure required.

This is correct without taking the clearance into account.

With the clearance added, the mean pressure would be slightly greater.

Or 2d, by means of the following table. The pressures per table are 15 pounds greater than the pressure that would be shown by a correct steam-gauge.

In the "Average Pressure" no deduction is made for back-pressure. Deducting, say six pounds for a condensing engine, we have the mean effective pressure.

For example it is required to find the power of an engine twenty inches diameter of cylinder, running at a piston speed of 600 feet per minute,

admitting steam of seventy pounds pressure, *by gauge*, to the cylinder, and cutting off at a quarter stroke. Add fifteen to seventy to find the absolute pressure; then against eighty-five and under a quarter cut-off the average pressure is found to be 50.65 pounds. This, less six pounds, 44.65 pounds is the mean effective pressure condensing.

MEAN PRESSURE OF STEAM AT DIFFERENT RATES OF EXPANSION.

Initial pressure in pounds per square inch.	Average pressure in pounds per square inch for the whole stroke. Points in the stroke at which steam is cut off.							
	$\frac{1}{10}$	$\frac{2}{10}$	$\frac{1}{4}$	$\frac{3}{10}$	$\frac{1}{3}$	$\frac{4}{10}$	$\frac{5}{10}$	$\frac{6}{10}$
40	13.21	20.87	23.86	26.22	29.67	30.66	33.86	36.14
45	14.86	23.48	26.84	29.73	33.38	34.89	38.09	40.66
50	16.51	26.09	29.82	33.03	37.07	38.32	42.32	45.18
55	18.12	28.57	32.86	36.67	40.83	42.08	46.47	49.91
60	19.79	31.02	35.77	39.23	44.49	45.98	50.73	54.41
65	21.49	33.84	38.71	42.98	48.35	49.89	54.98	58.96
70	23.18	36.43	41.73	46.22	52.00	53.52	59.07	63.25
75	24.72	39.00	44.82	49.53	55.73	57.36	63.38	68.00
80	26.41	41.66	47.75	52.88	59.41	61.13	67.47	72.45
85	28.02	44.08	50.65	56.14	63.17	65.00	71.84	77.00
90	29.69	46.89	53.73	59.43	66.94	68.83	76.00	81.47
95	31.34	49.37	56.62	62.81	70.52	72.68	80.16	86.00
100	33.03	52.00	59.63	66.01	74.23	76.47	84.37	90.57
110	36.41	57.46	65.87	72.97	81.82	84.12	93.00	99.95
120	39.77	62.50	71.84	79.44	89.10	91.98	101.44	108.63
130	43.01	67.85	77.49	86.00	96.55	99.47	110.00	117.89
140	46.32	73.00	83.38	92.51	104.00	107.21	118.36	126.77
150	49.73	78.11	89.12	99.04	111.63	114.56	126.89	136.00

CHAPTER IX.

SLIDE VALVES.

THIS is a branch of the subject deserving the special attention of the engineer.

Its importance in regard to the economical working of the steam engine cannot be overestimated.

The slide valve ordinarily used in steam engines, and the manner of its operation, are well known to nearly every practical mechanic and engineer. It will be remembered that the operations of admitting the fresh steam, and releasing the waste steam, are alternately performed by the same valve and the same motion.

The valve being made to slide backwards and forwards upon the face of the ports, opens and closes the several passages in their turn.

The two extreme ones, called the steam ports, communicate with each end of the cylinder.

The middle one is called the exhaust port, and its corresponding passage terminates in a pipe open to the atmosphere.

Steam is admitted freely into the steam chest from the boiler, and the valve is made of sufficient length to cover all the ports, when it is placed in the centre of the stroke.

When it is in this position no steam can enter the cylinder, but as the valve moves on one of the ports opens, the arrangement of the valve gearing being such that when the piston is ready to begin its stroke, the steam port begins to open.

During the stroke of the piston, the valve not only travels to the end of its stroke, but also returns to the point from whence it set out, and its continued motion in the same direction finally closes the port and prevents any further

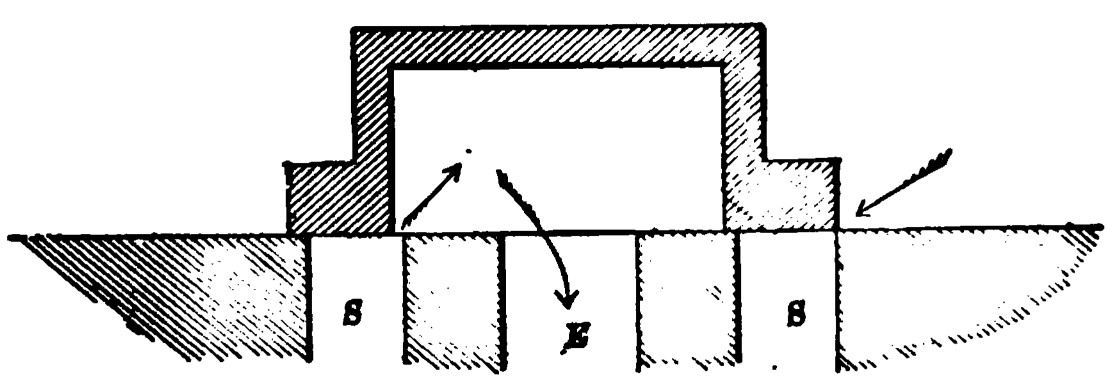

FIG. 9.

admission of steam. The steam has now done its work, and must be removed. In the middle of the valve a hollow chamber is formed of sufficient length to open between the ports.

As soon as the edge of this chamber passes the edge of the steam port, the pent-up steam finds vent and rushes through the exhaust port and escapes through the exhaust pipe into the atmosphere.

Now looking at Fig. 9 it will be noticed that the exhaust port opens when the steam port

closes, and that both happen just at the end of the stroke.

The perfection of a steam valve, other things being equal, consists in the degree of nicety with which its motion is timed relatively to the motion of the piston.

The functions of the piston are absolutely dependent upon the proper timing of the admission and release of the steam. A very slight and apparently trifling error in the adjustment produces a most serious effect upon the consumption of fuel. If from any cause the valve should open to admit steam for a fresh stroke before the preceeding stroke is finished, it opens too soon, and an unnecessary resistance to the piston is produced.

If, on the other hand, the valve should delay its opening until the piston had begun its return, it opens too late, because thus the steam has to fill uselessly the space left vacant, and hence a waste of steam, and a loss of power.

As far then as the admission of steam is concerned, it is a necessary condition that the *steam* ports should open neither before nor after, but at the precise moment when the stroke commences. Some engineers recommend giving the valve "lead" as it is termed, that is to say, setting it so as to open a little before the end of the stroke, but it is an open question whether the slightest advantage is gained by so doing to a greater ex-

tent than is necessary to compensate for any slackness or lost motion in the valve gearing, or for their expansion when heated by the steam, and $\frac{1}{16}$ of an inch is quite sufficient in a well-constructed engine.

It is also an open question whether it would not be better to bring the piston to a state of rest by the "compression" of the exhaust steam than by means of any lead to the steam valve at all.

Now the valve shown in Fig. 9, ratifies the conditions for the *admission* of the steam: it opens exactly at the right time, the steam begins to enter as the piston begins to move, as it follows it steadily and effectively throughout the stroke. Whatever *time* the piston takes for its journey, the steam is allowed as much time to follow it. At first the opening is small, but then the motion of the piston is comparatively slow, and therefore the supply keeps pace with the demand.

As respects the *release* of the steam when the stroke has been completed, the performance of this valve is altogether unsatisfactory, and here lurks the cause of the difference in the performance of the old and the later engines. But it may be said that the release does appear to take place at the right time, because it occurs just when the piston has finished its stroke, and if it were to occur before, a loss of power would ensue.

This is a very plausible view of the case, and the one which delayed for years the saving of fuel which has since been effected.

Sufficient attention was not bestowed upon what was going on in the cylinder, or upon the facts which might have indicated it; to fill and empty a cylinder full of steam being operations requiring *time.*

The time required for filling the cylinder with steam necessarily corresponds with the duration of the stroke, whatever its duration may be. But this cannot be the case as regards the second operation—emptying of the cylinder.

This ought to be performed in an *instant*, or otherwise the steam continues pent up when it ought to be liberated, when it ought to assume its minimum pressure—the pressure of the atmosphere—and exerts an injurious counter-pressure against the piston, tending to increase the resistance to be overcome. To effect the free and rapid discharge it is necessary not merely to open the communication to the exhaust pipe, but to open a wide passage and to have this done by the time the piston commences the return stroke.

The valve alluded to cannot accomplish this, its motion being gradual, not instantaneous, as it should be.

The passage only begins to open when the piston is in turn, and it is not wide open until the piston has travelled through one-tenth of its

entire stroke. The steam in the cylinder is restrained from escaping, being, as it were, wire-drawn in the passage out, and consequently takes considerable time to assume the pressure of the atmosphere. In the meanwhile the new stroke has begun and been partly completed, and so far the piston has had to contend with a resistance altogether illegitimate, a resistance in many cases—especially at high speeds—nearly equal to all the other resistances put together. It was not until the year 1838 that the true cause of the trouble was suspected, and a remedy applied.

It had been thought before that time, that giving an engine lead tended to improve its speed when already running at a high rate.

This was attributed to the opening of the steam port being wide at the commencement of the stroke, thereby increasing the facility for the entrance of the steam in following up the piston.

Its true explanation was found to be the *earlier release of the waste steam* and consequent diminution of resistance. As sometimes three-eighths of an inch, or even one-half of an inch "lead" was given to high-speed engines, it was decided to try the effect of opening the exhaust passage earlier by the same amount, while the steam port should still be made to open only at the beginning of the stroke.

An engine was chosen for the experiment the

valve of which resembled Fig. 10, placing the valve on the ports so as to allow the exhaust passage to be three-eighths of an inch open, the steam port at the same time to be one-quarter of an inch open. This space, therefore, was closed by adding to the length of the valve at each end one-quarter of an inch.

The eccentric was, of course, shifted on the shaft to correspond with the alteration, and the engine with the altered valve (see Fig. 10) again set to work.

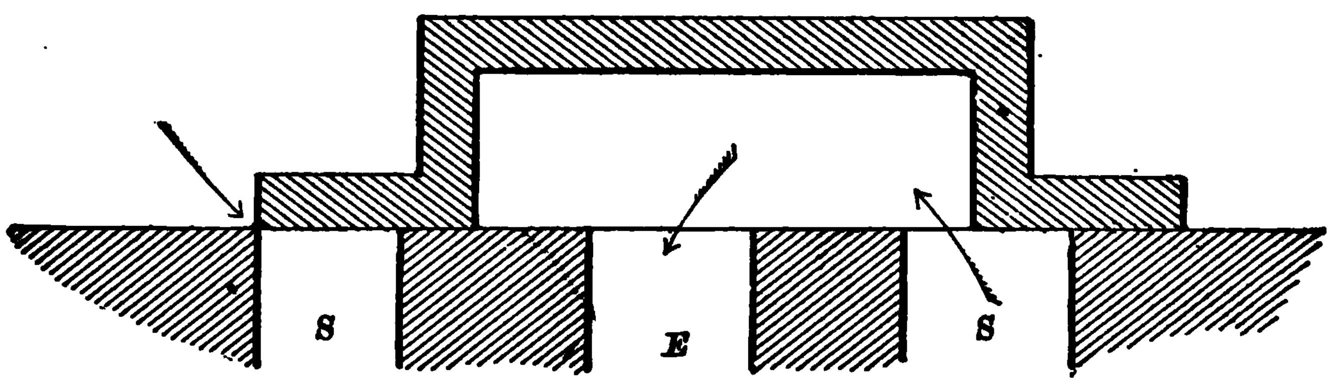

FIG. 10.

It is almost useless to say the saving of fuel was very great. The amount by which the valve at each end overlaps the steam ports (see Fig. 10), when placed exactly over them is technically termed the "lap." The lap of the valve being three-eighths of an inch, the exhaust passage was about three-eighths of an inch open when the stroke was finished.

LAP.

The importance of putting lap upon a slide valve will be better understood by noting what would happen without it.

If there were no lap the opening for steam would be represented by $O\ D$ (see Fig. 11), and the result would be that the steam port could only be perfectly closed at the precise instant when the valve was in the middle of its stroke, and moving rapidly. From what has been said it is apparent that a valve of this kind (one with-

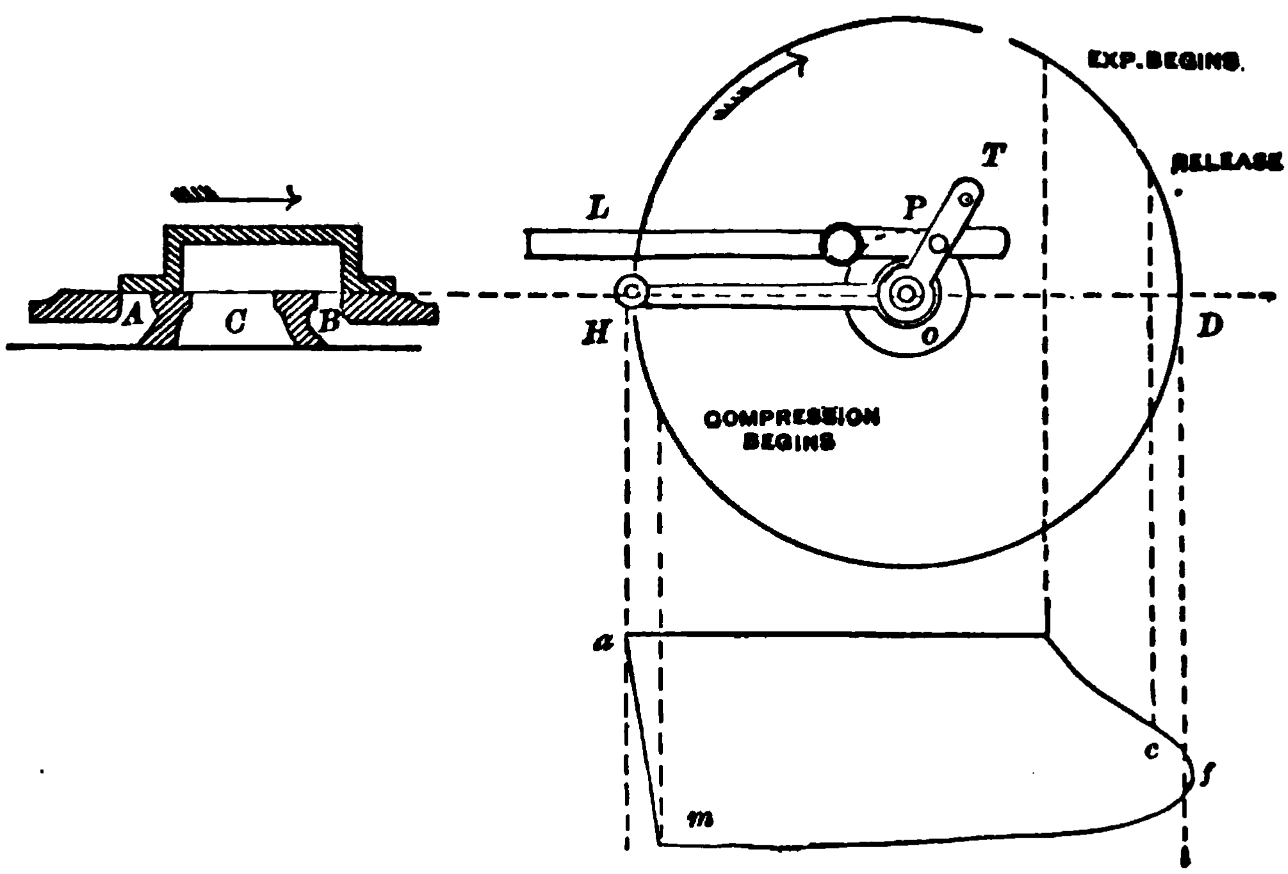

Fig. 11.

out lap) is unsuitable for an engine, the better plan being that the steam should be compressed or cushioned on one side of the piston, so as to assist in bringing it to a state of rest, and that the driving pressure on the opposite side should be relieved by opening a passage to the exhaust, or releasing the steam just before the stroke terminates.

This prevents the violent jerk and strain which would come upon the crank-pin if the steam were thrown with full force upon the piston when the crank is on the centre. The value of an indicator diagram in interpreting the action of a slide valve can now be made clear.

Referring to Fig. 11, it will be noticed that the moving parts are attached to a board carrying a sheet of paper on which the circles describing the centres of the crank-pin and centre of eccentric are recorded, and below this is a space for tracing the indicator diagram.

The crank and connecting rod which actuate the piston are at the back, but an index arm *O H* is placed in front, and moves with the crank, thereby transferring its apparent motion to the part where it can be seen. The eccentric is represented by an actual crank, *O P*, whose extreme end describes the smaller circle, and the rod *P L* carries on the motion of the valve.

The point *P* can be shifted along the arm *O T*, thereby varying the amount of travel of the sl de, and the length of the rod *P L* can also be ad usted. In this way the effect produced by any deviation from the proper length of the eccentric rod can be studied. We are now prepared to trace out the diagram as given by an indicator.

The crank being horizontal with the piston at the end of its stroke, the first thing to be done is to place the valve in the correct position for

admitting steam, by setting back $O\ P$ until the lap is allowed for.

The valve then opens, and if the pressure of the steam be sufficiantly maintained, the indicator pencil will trace the line $A\ B$.

When the crank gets to the end of the first dotted line, A is closed, expansion begins, the pressure falls, and we have the expansion $B\ C$.

At the point marked "release" the valve is moved so far to the left as to open a passage from A to C, and the release of the steam (exhaust) begins. The pressure falls from C to F, and continues very low till the point marked "compression" is reached, when B closes, and the steam in the corresponding end of the cylinder is "cushioned" so as to increase its pressure, the pencil rising from M to A when the double stroke has been completed. The object and effect of putting "lap" upon a valve are twofold:

1st, to give a free release to the exhaust steam, and 2d, to produce a fixed amount of expansion.

The "lead," of which mention has been made, is outside lead, that is, it relates to the *admission* of the steam; but of course, "lead" can be given to the exhaust side of the valve, in which case it is called *inside lead*.

The four principal points in the valve motion are: 1st, the admission of the steam; 2d, the cut-off; 3d, the release or exhaust; and, 4th, the

compression or cushioning of the steam behind the piston.

PISTON-VALVES.

Piston valves are merely circular slide valves, and all that is said about slide valves pertains to them as well as the common flat slide-valve in general use.

FIG. 12.—THE PISTON SLIDE-VALVE.

No system of balance slide-valves has yet been tried which has given entire satisfaction; some, indeed, produce more resistance than they have been designed to reduce, and the best cannot be depended on for any very long period when exposed to the pressure of high-temperature steam. The area of opening of port is restricted when only a common locomotive slide-valve is used, and its extensions magnify the evil which relief frames are supposed to cure. It has been observed that circular valves of the *mushroom* type do not work well in fast running engines, although they give a good opening to steam. To

combine the advantages of the two systems the piston valve is designed (Fig. 12). The port area is nearly three times that of a flat valve of the same dimension transversely, and the pressure on the sides due to the steam is nil. Essentially, the piston valve consists of two pistons, the face of each being equal in length to the bars of a locomotive slide valve, and connected by a rod. These pistons are fitted into a cylindrical chamber having ports corresponding to those in the cylinder face; the faces of the piston cover these ports, and have the same amount of lap, etc., as a common valve. Steam is admitted outside the pistons, and it exhausts from the cylinder into the space between them, and from there into the exhaust passage in the usual way.

When the pistons are sufficiently large they are connected by a pipe or hollow casting (as shown in Fig. 12), through which steam can pass from one end to the other; if this cannot be accomplished, the two ends of the valve-case are connected by a pipe cast with or connected to it.

Small engines, when fitted with such valves, have them in their simple form, the pistons being plain brass discs of the required thickness, generally cast in one piece. Such a form would suit all sizes of engines, if always working at full speed; but when standing or running slow, the leakage past the valve, when it was

worn, would soon be so considerable as to cause serious loss and make the engine very unhandy. To avoid this it is usual to pack the pistons much in the way that ordinary pistons are packed, except that the junk-rings and flanges are chamfered away, and the packing rings are made to project from them so as to allow free passage to the steam. The spring-rings are made of strong cast iron, or bronze with stiff cast iron lining rings inside them, and since, owing to the very slight velocity at which the valve moves, the wear is small, the rings should have very little *set* or spring. The liners in the valve-box are usually made of cast iron (hence bronze packing rings), fitted tightly in and secured by flanges.

There are *diagonal* bars across the ports to act as retaining guides to the packing rings; these bars are usually from ¾ to 1¼ broad, and take away about a third of the gross portway. The passage way around the liner must be so designed as to allow due area of section for the passage of steam; and to economize space, and reduce the clearance space, they are eccentric to the liner and valve. To avoid the chief defect in these valves—viz., large clearance space—the valve should be long so that its ports are nearly in line with those at the cylinder bore.

Piston valves are now becoming very general, and experience of them has given the necessary

confidence for their more extended use. For steam of a pressure over 100 lbs. they have become a necessity, and they may be used with advantage for the high-pressure cylinder for pressures down to 75 lbs. Some manufacturers use them for the low-pressure cylinder, but few engineers will care to incur the expense of them for a purpose where they are generally quite unnecessary. (*Seaton.*)

CHAPTER X.

PROPORTIONING PORTS AND SLIDE VALVES.

THE following is a simple, direct and comprehensive method of proportioning ports and slide valves for engines of any diameter of cylinder and variation of piston speed; also of getting the proper throw of valve, together with the time of opening and closing of exhaust.

In nearly all rules given, it is supposed that either the lap or travel of valve is first known. Now, the lap and travel of valve determine the port openings, and either of them *affects* the port openings. But they are of *first importance*, instead of being treated as secondary.

The first thing to be considered, in designing an engine, is the amount of work it is to perform; after this, the size of cylinder, together with the piston speed, steam pressure, and point of cut-off.

Suppose it is decided that the cylinder should be 16″ diam. by 30″ stroke, the piston to travel 500 ft. per minute=100 revolutions, and the cut-off to take place at ¾ stroke. This engine would give 90 H. P. As the port openings must be in proportion to the size of cylinder, together with the speed of piston, they should next

be found. We may get their area from Porter's rule—that steam, in entering a cylinder, should never be required to travel faster than 200 ft. per second.

In consideration that longer ports require larger valves and wider ports more travel, either of which creates more friction, it will be seen that port openings may be made too large as well as too small. We believe as good a rule as can be followed, for plain slide valve engines, is to suppose the piston to move at a uniform speed; the port to be at full opening all the time during admission, and velocity of steam 200 ft. per second.

The rule is:

Multiply the area of cylinder in inches by the speed of piston in feet per minute, and divide the product by 12000. This will give the area in square inches required for "live" steam,

$$\frac{500 \times 200}{12000} = 8.3 \text{ square inches.}$$

The exhaust steam should not have to travel faster than 110 ft. per minute. Make the steam port one inch wide, as shown in Fig. 13, and uncover it ¾" by the "wing" of the valve. This is the full linear opening for "live" steam while the exhaust has one inch linear opening. Divide the area 8.3 by ¾, which gives 11 inches as the length of port. The exhaust "throat" should be 2" wide, for the reason that the slide

the exhaust side and no lead on the steam side. Neither is the angularity of the connecting rod taken into account. But the latter has nothing to do with the travel of valves or the proportions of valves and seats, but should be considered in setting of valves.

By this simple process we have the following:

Cylinder 16″ x 30″	Throw of valve 3″
No. revolutions 100.	Lap outside ¾″
Cut-off, ¾ stroke.	Full port opening ¾″
Steam ports 1″ x 11″	Angular advance of eccentric 30°
Exhaust port 2″ x 11″	Linear advance ¾″
Length *ef* of face (Fig. 13) 2″	Exhaust closure 2.2″
Bars ¾″	

CHAPTER XI.

LINK MOTION VALVE GEAR.*

THE common form of motion employed to work the valves of a screw engine consists essentially of two eccentrics keyed on the crank-shaft in such a position relative to the crank that when one is operating on the valve, the engine will propel the ship ahead and is said to be in *head-gear*, and when the other, the engine will propel the ship stern first and is said to be in *stern-gear*, their rods being connected by a bar or bars on which is a block to which the valve spindle is attached. This bar connection is called the link, and is of such a form that by sliding it through the block, the *head* or *stern* eccentric may at pleasure be brought to operate on the valve.

In designing a link gear, the most important objects are to give the valve such motions as shall cause it to open to steam slightly before the piston is at the end of its stroke, the amount by which it is open at the end of the stroke, or commencement of the next stroke, being called

* Seaton.

the *lead* of the valve; to then open fully, and close at the required period of the stroke of the piston called *cut-off;* to confine the steam during the remaining portion of the stroke, so as to *expand* in the cylinder, and at or near the end of the stroke to allow the steam to escape from the cylinder, called *exhaust;* to close the port again before the end of the stroke, so that the piston compresses the steam remaining in the cylinder and port. These operations should be effected with the expenditure of as little power as possible, and with this end in view the motion of the link should be, as far as possible, limited to moving the valve only; consequently the link itself should have no sliding motion longitudinally, called *slotting motion*, in the block, but only the angular motion due to the two eccentrics. A perfect valve motion is such that the valve opens to steam *wide* immediately the crank has passed the *dead centre*, and remains open during the admission of steam, so that there is no wire-drawing; the valve closes suddenly, and remains closed during expansion; at the end of the stroke it opens wide to exhaust, and remains in that state during the whole period of exhaust, and at the end of it closes suddenly, and remains closed till opening again to steam. This is not obtainable with the ordinary link motion, nor to its full extent with any motion when one valve only is employed for both ends of the cyl-

inder, because the period of cut-off at one end does not, as a rule, correspond to the period of compression at the other end; but there are valve gears which have two periods of very quick motion and two of very slow in each cycle, which very closely fulfil the above conditions, and which will be noted later on.

Slot Link.—This, which is one of the oldest forms of link, is still retained by many engineers,

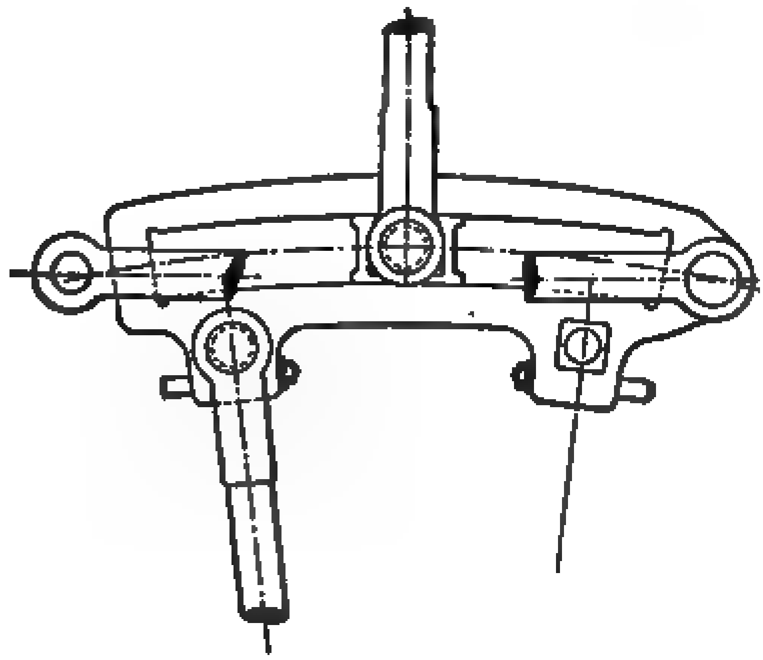

FIG. 15.

and is well adapted to the circumstances of several forms of engine, such as the oscillating paddle-wheel engine, and all engines in which there is not direct connection between the eccentric rods and the valve rod, and also in some of the horizontal engines when it is either impossible or inconvenient to have direct connection

Fig. 15 is an illustration of the ordinary slot link, having adjustment for the sliding block and eccentric-pin brasses. Locomotive engineers prefer, as a rule, to have the pin-holes fitted with hard bushes rather than adjustable brasses; but this opinion is not shared by marine engineers, and chiefly on the ground that in a foreign port it is seldom possible and never convenient to engage the services of workmen and tools to renew these bushes when so badly worn as to require renewal.

This kind of link is generally suspended from the end next the head-going eccentric rod, at a point in line with the arc through the block-pin; and if the pin in the lever, which operates on the link to reverse it, is placed in the proper position, there is very little slotting motion indeed when working in *head*-gear. The same remark applies to the position of the pin when in *stern*-gear, except that the amount of slotting motion is somewhat greater of necessity; but since a marine engine, as a rule, works but very little in *stern*-gear, and its efficiency there is of small consideration comparatively, this defect is of little moment.

Position of Suspension Pin.—To obtain the best position of lever pin, it is necessary to draw out the path of the centre of pin in the link end through one revolution of the engine when the link has *no slotting motion.* The path so found

is like an attenuated figure 8 in head-gear, and somewhat more pronounced in stern-gear. The arc of a circle of radius equal to the length of the suspension or bridle rods, is then drawn through each of these figures in such a way that there is the least possible deviation of the figure on either side; that is, the arc is the centre line of the figure, if the deviation on one side equals that on the other. The centres of the circles to which these arcs belong should be the centres of suspension of the bridle rods, or position of pin in reversing lever end. By drawing arcs of circles of radius equal to the length of the reversing lever from these two centres, the points of intersection are the two possible positions for the centre of weigh-shaft.

This same method of construction is suitable to all kinds of links, and for all positions of the point of suspension of the link.

When it is necessary that the motion shall be as efficient in *head*-gear as in *stern*-gear, the link should be suspended from a point in the line dividing it symmetrically, and by preference at the intersection of this line with the arc through the centre of block-pin, so that the centre of suspension is in line with the centre of block-pin when in *mid-gear.* When this is so, the pins for suspending the link are on side-plates bolted to the sides of the link.

The distance from centre to centre of eccen-

tric-rod pins should not be less than two and a half times the *throw* of the eccentrics, and is usually, when space permits, two and three quarters to three times. The *throw* of the eccentrics in this case is, of course, equal to the travel of the valve when in *full* gear.

Size of Slot Link.—Let D be the diameter of the valve spindle, from the calculation

$$D=\sqrt{\frac{L \times B \times p}{12{,}000}};$$

then,

Diameter of block-pin when overhung . . = D.
Diameter of block-pin when secured at both ends . . . . = 0.75 × D.
Diameter of eccentric-rod pins . . . . . . = 0.7 × D.
Diameter of suspension-rod pins . . . . . = 0.55 × D.
Diameter of suspension-rod pin when overhung . . . . = 0.75 × D.
Breadth of link . . . . = 0.8 to 0.9 × D.
Length of block . . . . = 1.8 to 1.6 × D.
Thickness of bars of link at middle . . . = 0.7 × D.
If a single suspension rod of round section, its diameter . . . . = 0.7 × D.
If two suspension rods of round section, their diameter . . . . = 0.55 + D.

The objections to the slot link are, that it is an expensive one to make, and that, owing to the eccentric pins and the block-pins being out of line, there is always an uneasy motion about the block-pin, and more slotting motion of the block. The former is valid, especially when the link is made of wrought iron, but when made of

cast steel it is not so expensive as some other forms. The uneasy motion is often due to bad design, for, when well designed and carefully hung, it will work very satisfactorily.

Single Bar Link.—This kind of link consists of a single solid bar, of rectangular section generally, and having the eccentric rods connected to each end, and a sliding block between, to which the valve spindle is connected. This form of link, although tried by more than one

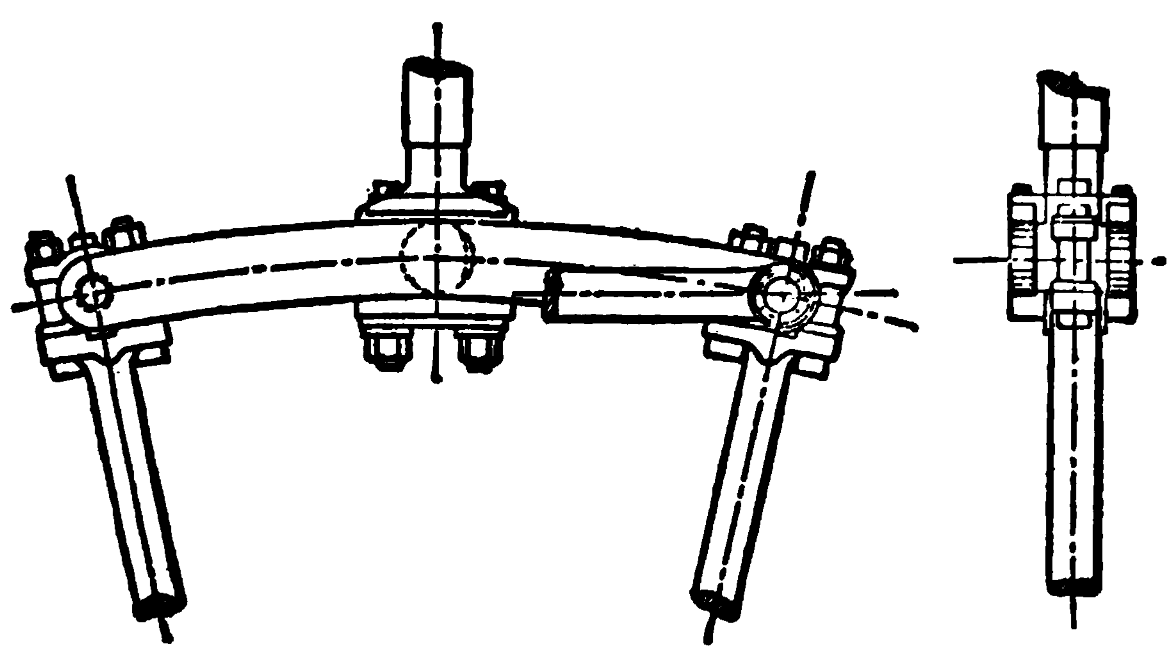

FIG. 16.

eminent firm of engineers, has gradually been dropped, notwithstanding the many ingenious elaborations devised to overcome its defects.

Double Bar Links.—There are two kinds of double bar links; one (Fig. 16) having the eccentric-rod ends, as well as the valve-spindle end between the bars, so that the travel of the valve is less than the throw of the eccentrics;

the other (Fig. 17) has the eccentric rods formed with fork ends, so as to connect to studs on the *outside* of the bars, and thus admits of the block sliding to the end of the link, so that the centres of the eccentric-rod ends and the block-pin are in line when in full gear.

The former plan is cheaper to make, is simpler in construction, and has fewer parts to get out of order and adjust; and when adjustment is required, it is easier to make, and there is less chance of its being done improperly. When in

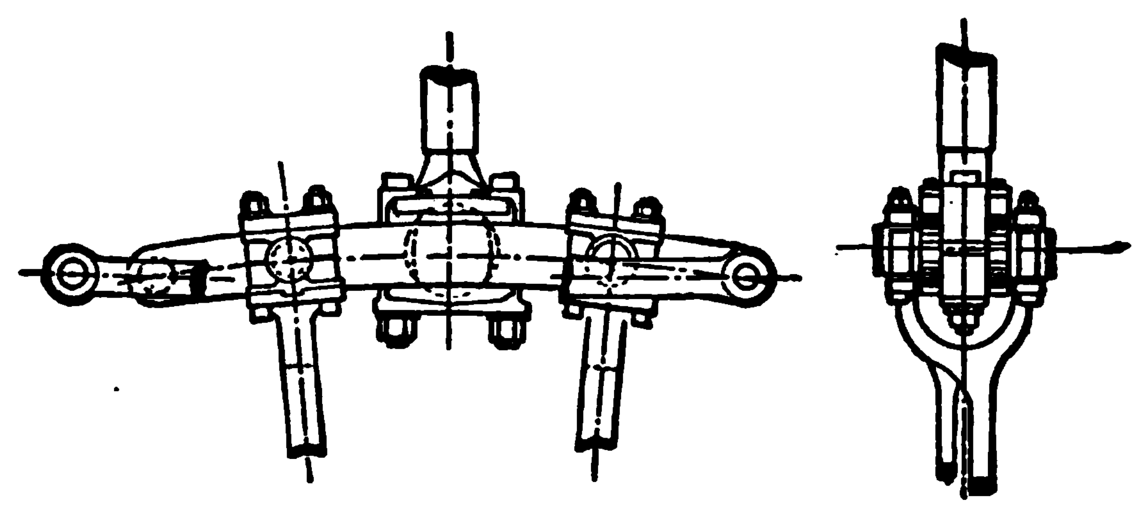

FIG. 17.

head-gear, part of the work of moving the valve is done by the *stern*-going eccentric, so that the wear is not limited to the one eccentric strap. When properly hung, the slotting motion is exceedingly small, and the valve motion is as perfect as with the other form. The objection to it is that the eccentrics are larger in diameter than those with the other links, and the links themselves are longer, and more space is required for the eccentric rods to move in.

Size of Bar Links.—Let D be the diameter of valve spindle, found as before.

Fig. 17, distance between centres of eccentric pins, 3 to 4 times throw of eccentrics.

Depth of bars	$= 1.25 \times D + \frac{3}{4}$ inch.
Thickness of bars	$= 0.5 \times D + \frac{1}{4}$ inch.
Length of sliding block	$= 2.5$ to $3 \times D$.
Diameter of eccentric-rod pins . . .	$= 0.8 \times D + \frac{1}{4}$ inch.
Diameter centre of sliding block . .	$= 1.3 \times D$.

Fig. 17, distance between eccentric rod pins $2\frac{1}{2}$ to $2\frac{3}{4}$ times throw of eccentrics.

Depth of bars	$= 1.25 \times D + \frac{1}{2}$ inch.
Thickness of bars	$= 0.5 \times D + \frac{1}{4}$ inch.
Length of sliding block	$= 2.5$ to $3 \times D$.
Diameter of eccentric-rod pins . . .	$= 0.75 \times D$.
Length of eccentric-rod pins	$=$

Diameter of eccentric bolts (top end), at bottom of thread $= 0.42 \times D$ when of iron; and $0.38 \times D$ when of steel.

These bars should be of a very good description of iron and case-hardened, or of steel; the latter is of course free from seaminess and stronger. The eccentric-rod pins of this kind of link (Fig. 17) are usually forged solid with the bars; but there is no absolute need of this, and it adds very much to the cost, both of manufacture and renewal when worn. Since the wear on these pins is limited to a very small portion of their circumference, it is not unusual to file away the parts which are not subject to wear, so as to admit of the brasses being closed when

worn. When loose pins are fitted, they should be steel, and hardened, so that all wear may come on the brasses which are capable of adjustment.

In another arrangement of bar link motion, the sliding block is divided, and on the outside, while the eccentric-rod ends are between the bars. This, while having some slight advantages, is on the whole very clumsy, and the block-pins wear badly; besides which, the link can be suspended only from the extreme end.

CHAPTER XII.

VALVE MOTION DIAGRAM.

The practical man has no time for formulas when he can travel (to him) a much straighter road to what he wants. The plan is simple, and any one can understand why he does this or that;

Fig. 18.

he knows he is right, and he has no conclusions to jump at.

The outside circle (Fig. 18) is the crank-pin

circle, the smaller one the eccentric's centre circle. The crank-pin circle is divided off into the inches of travel of piston, by a tram or pair of compasses set to represent the rod's length. *S S*, in Fig. 18, are the steam ports, *E* the exhaust port, and *X X* the bridges. Fig. 19, is a sectional elevation of the valve divided by the line *a d*. The valve drawing is cut out at the line *f e*, and placed on the line representing the valve-seat.

The laps and lead are laid off (from the pin *C*, the valve rod running directly from eccentric to

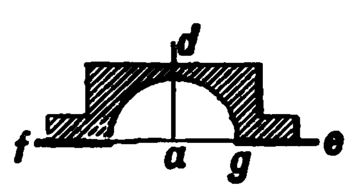

FIG. 19.

valve) *a b* from centre of circles, *a*. The line *b c* is erected perpendicular to centre line *F C*, and through where it cuts eccentric circle at *c* a line is drawn, as *a B*. This line is the centre line of eccentric, and the angle *C a B*—the angle formed by eccentric, and crank *c*. This angle is constant in whatever position the crank-pin may be.

Place the valve, Fig. 19, on the seat so that line *d a* coincides with line *c b*, and you have position of valve when crank is on centre. As the crank moves, in direction of arrow, from *C*, the valve moves from right to left, and back again, till right hand edge of valve comes to

edge of port, *z*. The steam is now cut off, and, to find position of crank-pin, notice that the line *a d* on valve cuts the centre line, *F C*, at *b*, which extended down to small circle, cuts it at *e*. The line *a e D* is now evidently the centre line of eccentric. Taking the distance, *B C*, in a pair of compasses, and measuring back from *D*, we find the point *A*, which is the position of the crank-pin when steam is cut off. The figures on the large circle give the number of inches traveled when this occurs. The valve now continues to move from left to right till edge of exhaust cavity *g*, comes to edge of port *h*, when the exhaust commences. The centre line on valve *d a* now cuts line *F C* at *a;* this, extended down, cuts eccentric circle at *m*. A line *a m j G*, now represents the centre line of eccentric; laying off the distance, *B C*, from *G*, we find the point *H* of crank-pin, when exhaust commences. The crank now travels on the lower half of its circle. The valve continues to travel from left to right, opening into the exhaust until the edge of exhaust cavity again comes to *h* (edge of steam port, the valve now traveling from right to left), the exhaust is closed and compression commences. This occurs, as before stated, when *g*, on valve, comes to *h*, on ports; the line *d a*, on valve, then cuts the line *F C* at *a*, and the line *a i K* represents centre line of eccentric. Lay off from *K* the distance *B C;*

and we find *I*, the position of crank when compression commences. Similar points on the opposition stroke can be found by commencing with crank-pin at *F* instead of *C*, noting that centre line of eccentric is down instead of up. If the eccentric's motion is transferred to valve through a rocker shaft, the lap and lead must be laid off *toward* the crank-pin instead of from it.

These points, we assume, are understood by the reader, as it would require too much space to explain them all. The irregularity of the valve's motion (more expansion occurring on one end than the other) can be ascertained by this diagram. Unless the eccentric rod is *very* short, no notice may be taken of its irregularity due to radius (shown in line *b c*, being struck from length of eccentric rod), as it is so little that, generally, a straight line, as *b c*, will suffice for practical purposes.

We trust that this may be of use to some who have worried their brains with impracticable formulas and diagrams, produced by men more scientific than practical, and which resemble puzzles.

MOTION CURVES.

The laying off of motion curves presents to the eye *all* of the movements of the valve at a glance. Fig. 20 shows the diagram after it is completed, ready to file away for future reference. Elevations of the ports are laid out to a scale and in

length equal to the stroke of the piston, with the inches marked as shown. We start with the valve *V*, as shown at the commencement of the stroke, and, by means of Fig. 20, get the

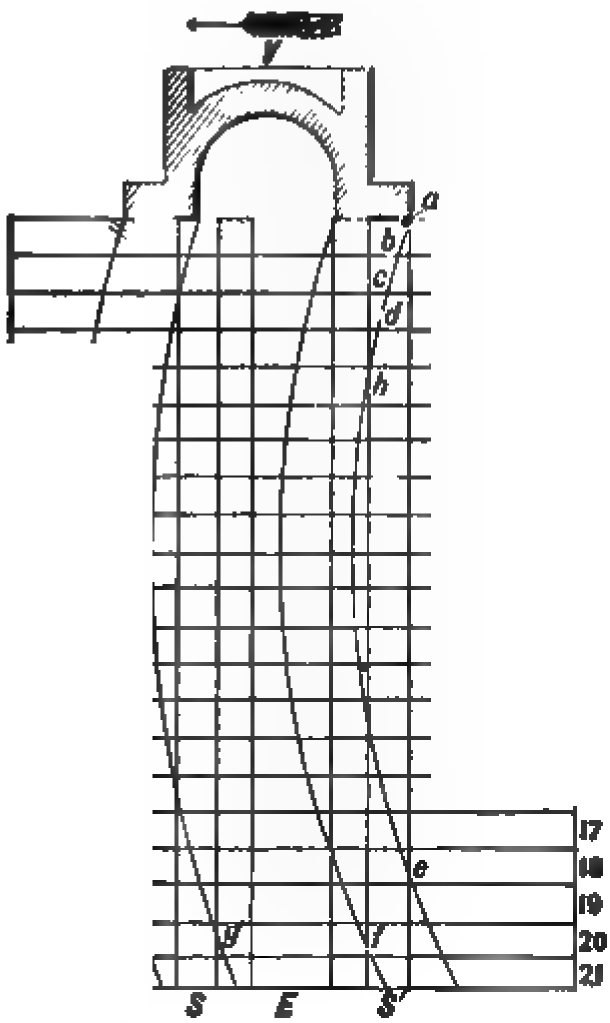

FIG. 20.

position of the valve for several positions of the piston, or for each inch, as shown at *a*, *b*, *c*, *d*. When all of the positions are obtained and jotted down on the port elevations, a curve that cuts them all is drawn, as are similar curves for

the other edges of the valve. The opposite curves can also be laid down, and the movements of the valve for the opposite stroke obtained. In the diagram shown, we see that the port S′ was wide open at *h*, or 4½ inches of piston travel; that steam was cut off at *c*, 18 inches; exhaust opened at *f*, 19½ inches, and compression commenced at *g*, 19½ inches also, the valve being "line and line." Other points can be obtained by inspection, and the diagram forms valuable data for the builder and user.

TABLE BY WHICH TO FIND THE RELATIVE STATE OF PISTON AND EXHAUST AFTER EXPANSION.

	Steam cut off at ⅓ from the end of the stroke.		Steam cut off at ¼ from the end of the stroke.		Steam cut off at ⅙ from the end of the stroke.		Steam cut off at ⅛ from the end of the stroke.	
Cover on the exhausting side of the valve in parts of the length of its stroke.	Distance of the piston from the end of its stroke, when the exhausting-port before it is shut (in parts of the stroke).	Distance of the piston from the end of its stroke, when the exhausting-port behind it is opened (in parts of the stroke.)	Distance of the piston from the end of its stroke, when the exhausting-port before it is shut (in parts of the stroke).	Distance of the piston from the end of its stroke, when the exhausting-port behind it is opened (in parts of the stroke).	Distance of the piston from the end of its stroke, when the exhausting-port before it is shut (in parts of the stroke).	Distance of the piston from the end of its stroke, when the exhausting-port behind it is opened (in parts of the stroke).	Distance of the piston from the end of its stroke, when the exhausting-port before it is shut (in parts of the stroke).	Distance of the piston from the end of its stroke, when the exhausting-port behind it is opened (in parts of the stroke).
⅛th	.178	.033	.143	.019	.109	.008	.093	.004
$\frac{1}{16}$th	.130	.060	.100	.040	.171	.022	.058	.015
$\frac{1}{32}$d	.113	.073	.085	.051	.058	.033	.043	.023
0	.029	.092	.067	.067	.043	.043	.038	.033

Example from the Table.—Suppose an engine with a stroke of six feet, or 72 inches, and the steam cut off when the piston is one-third from the end of its stroke, the cover on the exhaust side of the valve being 1-32d of its stroke, the relative position will be the following:

In a line with 1-32d and under 1-3d is .113 and .073; therefore $.113 \times 72 = 8.136$ inches the piston is from the end of the stroke, when the exhausting-port before the piston is shut, and $.073 \times 72 = 5.256$ inches the piston is from the end of the stroke, when the exhausting-port behind it is open.

Table by which to ascertain the amount of lap necessary on the steam side of a slide-valve to cut the steam off at various fractional parts of the stroke.

To cut the steam off, after the piston has passed through						
$\frac{1}{2}$	$\frac{7}{12}$	$\frac{2}{3}$	$\frac{3}{4}$	$\frac{5}{6}$	$\frac{7}{8}$	$\frac{11}{12}$
of its stroke. Multiply the given stroke of the valve by						
.354	.323	.289	.250	.204	.177	.144
and the product is the lap of the valve in terms of the stroke.						

Example.—Required the lap necessary to cut the steam off at the end of five-sixths of the stroke, the stroke of the valve being twelve inches, and without lead. $.204 \times 12 = 2.448$ inches.

As lead is not taken into account because of different quantities being required to different applications of the steam-engine, subtract from

the lap half the lead; the remainder is the lap required. Thus, suppose the lead equal $.25 \div 2 = .125$ and $2.448 - .125 = 2.323$ inches, the lap with one-fourth inch of lead as given.

CHAPTER XIII.

HOW TO SET A SLIDE VALVE.

It is first necessary to find the two centres for the cross-head. *A*, in Fig. 21, is the rim of the balance wheel, and *B*, a piece of wood of a length to reach from the floor or foundation to about the height of the centre of the shaft.

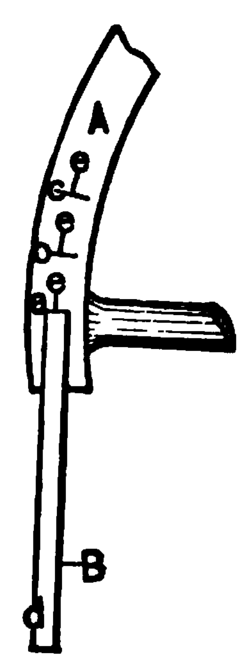

FIG. 21.

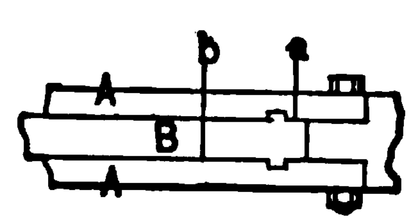

FIG. 22.

On the guides *A*, Fig. 22, draw a line *a*, at or a trifle beyond the travel point of the cross-head. Measure off a short distance (2, 3, 4 or 5 in.) on the guide from line *a*, and draw the line *b*. Place the cross-head as near on the centre as can be done with the eye, and turn the shaft, the

crank-pin travelling up till the cross-head arrives flush with line *b*, as shown, *B* being the cross-head. Then with the stick resting on the floor or foundation, and against side of balance-wheel, make a mark *a*, Fig. 21, as shown. Now return the cross-head to the centre, turning the shaft in an opposite direction to that in which it was first moved, till the pin passes the centre and travels down, and the cross-head has again arrived at line *b*, as shown in Fig. 22, the stick *B* remaining in the same position, another line, *c*, may now be drawn flush with the end of the stick.

The centre between *a* and *c* may now be found and drawn, as *b*. Now, when the line *b* is brought flush with the end of the stick by turning the shaft, the cross-head will be on the centre. A similar operation for the other centre will also ascertain it.

The length of the eccentric rod is next in order, and should be lengthened or shortened as the case may be, till the valve opens one port about or nearly as far as the other.

While doing this the position of the eccentric makes no difference, and may, in fact, be fastened anywhere.

Having approximated the correct length of the rod, place the engine on the center (either).

The direction in which the engine is intended to run being known, the position of the eccen-

tric must be ascertained. If the engine has a rocker-shaft, the full part or belly of the eccentric will follow the crank-pin; if there is no rocker-shaft, the full part of the eccentric will lead the crank-pin, being in either case nearly at right angles with the pin. The engine being on the centre, as in Fig. 23, and having no rocker-shaft, we will suppose it is to run in the direction of the arrow. As the eccentric in this case leads the pin, it will be in the position of *A*.

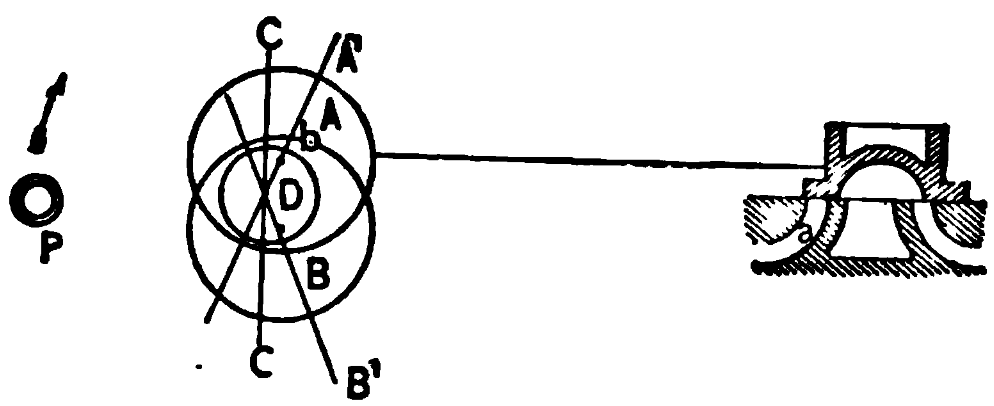

FIG. 23.

Now adjust it till the required lead appears at *a*, fasten it, and try the lead when the engine is on the other centre.

Whatever difference there is in the lead, lengthen or shorten the rod till the lead is equally divided on both ends, not moving the eccentric till this is accomplished, and when it is, shift the eccentric to the crank if there is too much lead, and from it, if there is not enough.

The centre of the eccentric *b*, will then be the lap and lead from the centre line *c c*.

If the engine was to run in the opposite direc-

tion, the eccentric would occupy the position of *B*. That the position *A* is the correct one, is evident from inspection; for imagining the pin *p* to commence and travel in the direction of the arrow, it is evident that the eccentric *A* will drive the valve from left to right, opening the port *a* as required.

If we connect the valve-stem to eccentric *B*, and suppose the pin to still travel in the direction of the arrow, the eccentric would drive the valve from right to left, and close the port *a*, and it is plain that *B* is not in the correct position for the direction of the arrow.

However, if the engine is to travel in the opposite direction, it at once becomes evident that *B* is in the correct position for that direction. In either case, the valve occupies the same position when the pin *p* is on the centre.

Locomotive engineers find in this fact an explanation for setting a slipped eccentric by throwing the reverse lever in the opposite direction to the slipped eccentric, marking the valve-stem, throwing the lever in the motion for the slipped eccentric, and shifting the eccentric till the mark on the valve-stem reappears in the same relative position as when it was made, as the valve occupies the same position when the engine is on the centre, whether the engine runs forward or backward.

By examining Fig. 24 it will be seen that the

rocker-shaft makes all the difference in the world; in fact, the eccentric stands opposite to what it would without the rocker-shaft. When the engine runs in the direction of the arrow, the eccentric pulls the eccentric rod in the direction of the arrow under it, and drives the valve-stem in the direction of its arrow from the left to the right, as is required. The rocker-shaft just reverses the order of things, that is, it causes the

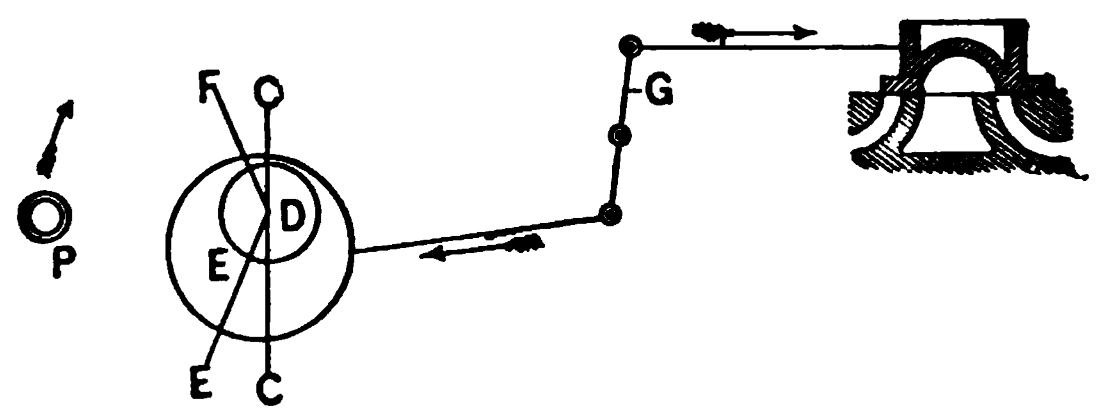

FIG. 24.

eccentric to be the lap and lead *nearer* to the pin from a centre line *c c*, and also follow the pin; while without the rocker-shaft, the eccentric is as seen in Fig. 23, the lap and lead further from the pin, from the line *c c*, and it also *leads* the pin.

When the rocker-shaft is used, it should stand at right-angles with the valve-stem, when the valve is over the centre of the seat.

CHAPTER XIV.

ENGINE CONSTRUCTION*—DETAILS.

The Cylinder.

THE Cylinder Barrel should be stiffened by external flanges or webs, at about 12 times the thickness of metal apart; these webs should be $1.5 \times f$ thick, and stand at least $0.75 \times f$ beyond the surface of the cylinder. Some engineers, however, prefer to do without these stiffening webs, and make the cylinder somewhat thicker instead.

The following rules are for the cylinder and its connections:

D is the diameter of the cylinder in inches.

p, the load on the safety-valves in lbs. per square inch.

p_1, the absolute pressure of steam in the boiler.

f, a constant multiplier = thickness of barrel + .25 inch.

Thickness of metal of cylinder barrel or liner, not to be less than $p \times D \div 3000$ when of cast iron.

Thickness of cylinder-barrel $= \dfrac{p \times D}{5000} + 0.6$ inch.

Thickness of liner $= 1.1 \times f$.

Thickness of liner when of steel $p \times D \div 6000 + 0.5$.

Thickness of metal of steam ports $= 0.6 \times f$.

Thickness of metal of valve-box sides $= 0.65 \times f$.

Thickness of metal of valve-box covers $= 0.7 \times f$.

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*Seaton.

Thickness of metal of cylinder bottom $= 1.1 \times f$, if single thickness.

Thickness of metal of cylinder bottom $= 0.65 \times f$, if double thickness,

Thickness of metal of cylinder covers $= 1.0 \times f$, if single thickness.

Thickness of metal of cylinder covers $= 0.6 \times f$, if double thickness.

Thickness of cylinder flange $= 1.4 \times f$.

Thickness of cylinder cover flange $= 1.3 \times f$.

Thickness of cylinder valve-box flange $= 1.0 \times f$.

Thickness of cylinder door flange $= 0.9 \times f$.

Thickness of cylinder face over ports $= 1.2 \times f$.

Thickness of cylinder face over ports $= 1.0 \times f$, when there is a false face.

Thickness of cylinder false face over ports $= 0.8 \times f$, when cast iron.

Thickness of cylinder false face over ports $= 0.6 \times f$, when steel or bronze.

Main Steam Pipe.—The main steam pipe, which supplies a cylinder with steam, should be of such a size that the mean velocity of flow through it does not exceed 8,000 feet per minute. When this is not exceeded, the loss of pressure between the boiler and the valve-chest is very slight indeed. If, however, the valve-chest is large, and the cut-off in the cylinder is before half stroke, the area of transverse section of this pipe may be smaller than given by the above rule, inasmuch as the piston speed is below the mean velocity at the early part of the stroke, and the space in the steam-chest acts as a reservoir for steam, so as to keep up a steady supply during admission.

Taking 8,100 feet as the mean velocity, *S* the mean speed of piston in feet per minute, and *D* the diameter of the cylinder, then,

$$\text{Diameter of main steam pipe} = \sqrt{\frac{D^2 \times S}{8100}} = \frac{D}{90} \sqrt{S.}$$

Example.—To find the diameter of the main steam pipe to a cylinder 45 inches diameter and 60 inches stroke, the revolutions at full speed to be 60 per minute.

Here $S = 2 \times 5 \times 60 = 600$, and $D = 45$ inches.

Therefore,

$$\text{Diameter of main steam pipe} = \frac{45}{90} \sqrt{600} = 12.25 \text{ inches.}$$

Cylinder Liner.—In order that a suitable material may be supplied to resist the rubbing action of the piston without wearing away, and one that shall be capable of taking and retaining a polished surface, so as to minimize the friction of the piston, an inner bush or false barrel is fitted, usually called the *cylinder liner.* This liner should be made of a hard, close-grained metal having considerable strength, but not so hard as to resist the action of a cutting tool or file; it should also be such that the expansion caused by heat is very nearly the same as the cast iron of which the cylinder itself is made. It is usual to make these liners of cast iron, strengthened, closed and hardened by mixing with it certain kinds of pig iron, pressed or other

equally good steel, hammered out to the proper size for boring; and some engineers use cast steel. Although the compressed steel gives good results, it can be equalled by the specially-made cast iron, so far as good wearing is concerned, but, of course, it far exceeds cast iron in strength; this latter quality is necessary to a higher degree for the horizontal engine than for the vertical engine.

In the merchant service, with the vertical engine, the cast iron liner does exceedingly well, and is not likely to be superseded by steel, even

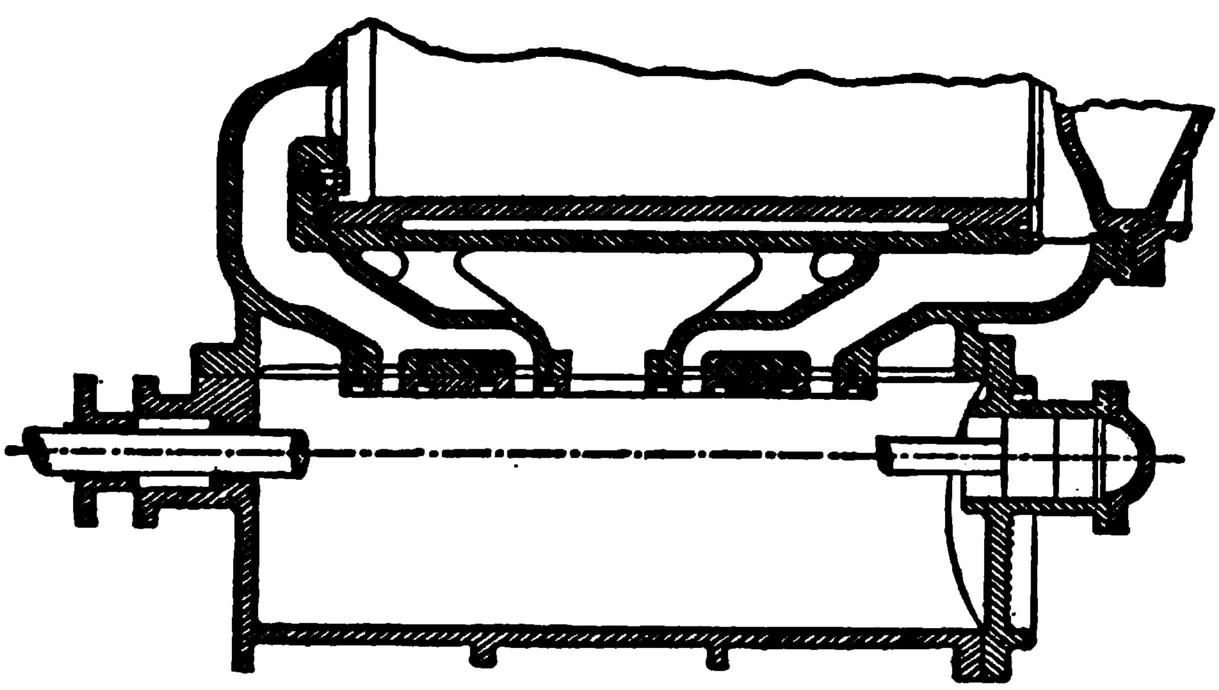

FIG. 25.—SECTION THROUGH CYLINDER.

if this material can be manufactured much cheaper than at present. Liners are usually made with an inside flange at the bottom end (Fig. 25), which fits into a recess in the cylinder end, and is secured there by *screw-bolts*. The upper end is turned for a few inches, so as to fit

tightly into the cylinder shell at that part. The joint at the cylinder bottom is made with red lead paint, while leakage between the liner and the cylinder shell is prevented at the other end by stuffing a few rounds of gasket, rope, or Tuck's packing into a recess formed for that purpose, and preventing it from coming out by securing a flat wrought-iron ring *to the liner* so as to cover the packing. Sometimes in lieu of a stuffing-box, the outer edge of the liner and the edge of the turned part of the cylinder shell are chamfered so as to form a groove; into this groove a turn of Tuck's packing or abestos rope is pressed with a ring as before. Some engineers, preferring to rely on metallic contact, turn a slight recess instead of chamfering the edge of the liner, and *caulk into it* a strip of soft copper. The liners are sometimes secured without a flange at the bottom, by screwing studs through the cylinder shell and liner, and making the ends steam-tight as before.

The space between the liner and shell should not be less than 1 inch, and may be filled with steam so as to heat the steam during expansion. If the cylinder has to be jacketed, this is really a better plan of doing it than by casting the cylinder and inner cylinder together, as was very generally done formerly.

Pistons.—Pistons of marine engines above 12 inches diameter for a high-pressure, and 20 inches

diameter for a low-pressure cylinder, are usually made cellular—that is, with two thicknesses of metal stiffened or connected by ribs and webs, and either by the thickness of metal or by the depth of body made strong enough structurally to safely withstand, not only the steam pressure exerted on it and transmitted to the rod, but also the shocks to which it is liable when priming occurs.

The piston body must be so designed, too, that it may be safely cast, for in the early days of large pistons it was not at all an uncommon thing for a piston to break in cooling, or mysteriously afterwards. For this reason any rules must of necessity be empirical which set out the thickness of metal of the different parts of the body; but care must always be exercised that no one part is too small for the strains to which it is subject. For example, there must be sufficient metal in the immediate neighborhood of the piston-rod boss to resist the tendency to force out this part by shearing the metal. Again, the piston may be taken as consisting of a number of sectors, and by considering one of such small sectors loaded with the pressure on its area at the centre of gravity of its figure, the bending moment at any section may be found, and the thickness of metal tried whether it be sufficient for the purpose.

For the section of an ordinary piston having a

single rod, the following table gives the multipliers for obtaining the thickness of metal and sizes of the different parts.

Details of Construction of the Ordinary Piston.—Let D be the diameter of the piston in inches, p the effective pressure per square inch on it, x a constant multiplier, found as follows:—

$$x = \frac{D}{50} \times \sqrt{p} + 1.$$

The thickness of front of piston near the boss	$= 0.2 \times x.$
" " " " rim	$= 0.17 \times x.$
" back of piston	$= 0.18 \times x.$
" boss around the rod	$= 0.3 \times x.$
" flange inside packing-ring	$= 0.23 \times x.$
" flange at edge	$= 0.25 \times x.$
" packing-ring	$= 0.15 \times x.$
" junk-ring at edge	$= 0.23 \times x.$
" " inside packing-ring	$= 0.21 \times x.$
" " at bolt holes	$= 0.35 \times x.$
" metal around piston edge	$= 0.25 \times x.$
The breadth of packing-ring	$= 0.63 \times x.$
The depth of piston at centre	$= 1.4 \times x.$
The lap of junk-ring on the piston	$= 0.45 \times x.$
The space between piston body and packing-ring	$= 0.3 \times x.$
The diameter of junk-ring bolts	$= 0.1 \times x + 0.25$ in.
The pitch of junk-ring bolts	= 10 diameters.
The number of webs in the piston	$= \frac{D + 20}{12}.$
The thickness of webs in the piston	$= 0.18 \times x.$

When made of exceptionally good metal, at least twice melted, the thicknesses may be as

much as 20 per cent. less than given by the rules; but, on the other hand, if made of other than really good metal they should be thicker. The piston should be made of good metal always, and for fast-running engines it is better made of steel. The packing ring is sometimes made much thicker in the part opposite the cut than given above, in order to have sufficient elasticity of itself to press steam-tight against the cylinder; but it is better to let the springs perform their function wholly, and leave the ring to act only as the packing.

PISTON RINGS AND SPRINGS.

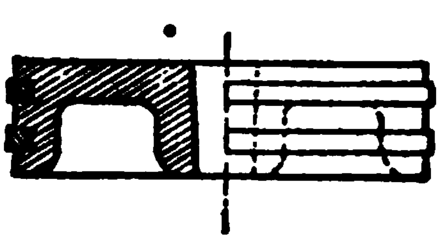

FIG. 26.

Ramsbottom's Rings (Fig. 26).—Mr. Ramsbottom was the first to pack pistons by one or more narrow metal rings, turned somewhat larger in external diameter than that of the cylinder bore, and which, after being cut across so as to be capable of being compressed to suit the bore of the cylinder, are fitted into recesses turned in the piston edge. The rings fit accurately into these recesses, and as they are so placed that no two of the joints are in a line, the piston is practically steam-tight, and works very

well in locomotives and other quick-working engines of small size; but for large engines, and engines undergoing the same vicissitudes as those on shipboard, there is an objection to this form of piston. It will be seen that the rings cannot be removed without drawing the piston, and that there is no means of preventing steam from passing where the spring is cut across, besides which the rubbing surface is very small, and the spring is always exerting its maximum effort. The first of these objections is overcome by fitting a junk-ring, having cast with it a

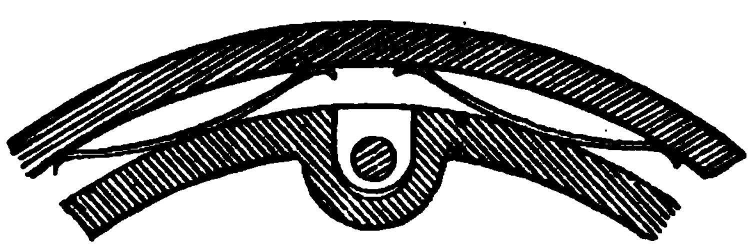

FIG. 27.

spigot or ring, which goes down into the recess around the piston for the packing ring, and made steam-tight; into grooves turned in the outer surface of this spigot the Ramsbottom rings are fitted.

For small engines these rings are made of steel; for such engines as may be standing unused for many days, some engineers prefer to fit hard brass rings. When for larger engines where the section may be three-quarters of an inch square and upwards, the rings are better of tough and hard cast iron.

Common Piston Rings (Fig. 27) consist only of a single hoop made of very tough, close-grained, cast iron, made on the same principle as the Ramsbottom rings, but fitted between the piston flange and the junk-ring, so as to be free to move laterally steam-tight. This packing ring is usually turned to a diameter about 1 per cent. in excess of that of the cylinder, and cut across diagonally.

The ring is then fitted to the piston flange steam-tight by scraping both surfaces; the ring is raised by interposing very thin pieces of paper between it and the flange, and the junk-ring is then fitted steam-tight to the piston and packing ring by scraping, etc. Some makers of pistons profess to turn the piston and rings so accurately as to require no scraping, but it is doubtful if there is economy in the practice if carried to the perfection professed; other engineers prefer to grind the rings tight after coming from the lathe. In whatever way the object is attained is of small moment compared with the necessity of having the ring *perfectly* steam-tight between the flange and follower.

Piston Springs.—When the piston is of comparatively small diameter, the elasticity of the packing ring itself is sufficient to keep it steam-tight against the cylinder sides for a very considerable time after it is fitted; and even larger rings may be made of sufficient strength to do

this, but they would then be open to the same objection as raised against the Ramsbottom rings. The old method of pressing the ring out by means of dished springs or coach-springs, as shown in Fig. 27, is now seldom used in new engines; the objections to it are the uneven and unknown pressure exerted, and the reaction of the piston itself, from the fact of the springs pressing on it. It was a very difficult thing to so set every spring that the pressure on the ring was uniform; and the range of action of this form of spring is very limited, so that although the ring might be very tight when first fitted, after a few days' running it would be passing steam. The surface exposed to pressure too was small, and the springs were apt to bed themselves into the ring, and in doing so wear through their curved ends. These defects were partially remedied by adding to each one or more subsidiary springs on the principle of coach springs; but that only tended to aggravate the other evil spoken of—viz., the reaction of the piston itself.

When a piston is moving through its course, and guided therein by the rod at one end and the tail rod (or back guides in case of a horizontal engine) at the other, it should be quite free laterally from the packing ring, which may follow its course freely. When the bore of the cylinder is quite true, and its axis coincides

with the line of motion of the piston centre, it is of no consequence if the springs do bear on the piston; but if the cylinder wears somewhat out of truth in either direction, it is important that the spring-ring shall follow the sides of the cylinder freely; it cannot do this if the springs react from the piston body.

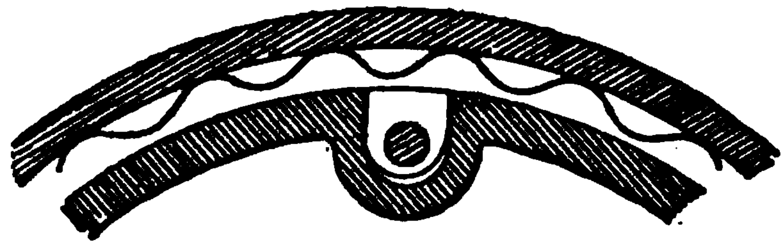

FIG. 28.

Cameron's Patent.—Fig. 28 shows a piston ring pressed out with a corrugated ribbon of steel; the lateral pressure here is obtained by the resistance of the spring to being bent into a circle, and by the pressure exerted by the corrugations when the ends of the spring are pressed apart. This spring exerts an almost uniform lateral pressure on the packing ring without touching the body of the piston, and by making the packing ring comparatively thin, it will adapt itself to the shape of the cylinder when worn. The pressure on the ring can also be easily and nicely adjusted by packing pieces between the ends of the spring. One great advantage of this spring is that it can be fitted to any piston without condemning any of the parts beyond the springs.

PISTON RODS.

The following rules will give results sufficiently accurate for all practical purposes:

$$\text{Diameter of piston-rod} = \frac{\text{Diameter of Cylinder}}{F}\sqrt{p}.$$

The following are the values of F:—

Naval engines, direct-acting		F = 60.
"	return connecting rod, 2 rods	F = 80.
Mercantile ordinary stroke, direct-acting		F = 50.
"	long " "	F = 48.
"	very long " "	F = 45.
"	medium stroke, oscillating	F = 45.

GUIDE BLOCKS AND SLIDES.

Surface of Guide-block.—The area of the guide-block, or slipper-surface, on which the thrust is taken, should in no case be less than will admit of a pressure of 400 lbs. on the square inch; and for good working those surfaces which take the thrust when going *ahead* should be sufficiently large to prevent the maximum pressure per square inch exceeding 100 lbs. per square inch. When the surfaces are kept well lubricated this allowance may be exceeded, but the reduction in surface should be effected by making shallow grooves and recesses in the face of the slipper, in which the lubricant can lodge and impart itself to the guide as it is carried along. A good method of carrying this into effect is to provide a surface calculated on the allowance of 100 lbs. per square inch, and by cross-planing

FIG. 29. FIG. 30.

CONNECTING-RODS.

TO FACE PAGE 154

so as to leave shallow recesses about $\frac{1}{16}$ inch deep, reduce the actual surface which touches the guide to about $\frac{5}{8}$ of the original area; there will be then strips across the slipper $1\frac{1}{4}$ inches wide, with depressions between them $\frac{3}{4}$ inch wide, filled with grease.

Cast iron, hard and close grained, is the best material for the guide-plates; its surface, after a few hours' work, becomes exceedingly hard and highly polished, and offers very little resistance to the slipper or guide-block. So long as this hard skin remains intact, no trouble will be experienced, but if abrasion takes place from heating or other cause, it rarely works well after, and should be at once planed afresh.

Guide Plates.—In order to have a sound and hard surface for the piston-rod slides to work on, the guide-plates should be separate from and secured to the columns; when so fitted they also admit of adjustment, and may be cast hollow, so as to permit of a flow of cooling water through them. This is especially needful for large quick-running engines, where the speed of piston is very high, and any want of lubrication would soon cause most serious damage. Cast iron when once worn smooth gives a splendid surface for a slide; but if by any mischance this surface suffers a little abrasion, it is most difficult to get right again, and will seldom work well again until it is.

The face of the guide plates should have good oil courses cut on it, so that the lubricant is well distributed, and they should be cut deep enough to prevent their being choked with the gluey deposit from the oil. The piston-rod slide should always be provided with a comb, which will carry the lubricant from the drip-boxes, and spread it over the face of the guide.

Area of steam ports and of section through passages

$$= \frac{\text{Area of piston} \times \text{speed of piston}}{6000}$$

$$= \frac{(\text{Diameter of cylinder})^2 \times \text{speed of piston}}{7636}$$

Opening of Port to Steam.—It is advisable so to design the valve, &c., that the opening for admission of steam to the cylinder is sufficient to avoid any serious loss by "wire-drawing;" but in actual practice, unless special gearing is designed so as to give a quick motion to the valve at the instant of cut-off, there is very considerable loss of pressure shown on the indicator-diagram; and, what is worse still, from deficient opening, the loss is generally not limited to the period of cut-off, but during the whole time of admission. The ordinary valve-gears do not give that quick motion, either at opening or at cut-off, which is such a desideratum. Separate expansion valves and special valve-gearings admit of such a motion, and consequently the

opening to steam with them may be smaller than when cut-off is effected by the ordinary slide-valve and link-motion.

Hence, when only common valves and gear are to be used, the area of opening to steam when at its greatest should be such that the mean velocity of flow does not exceed 10,000 feet per minute. When expansion valves, or special valve-gearing, is used, the mean velocity may be assumed at 12,000 feet, although it is better to give such an amount of opening, when possible, that the velocity shall not exceed 10,000 feet. In actual practice the amount of opening is often much less than that given by the above rules, but it always results in loss of pressure in the cylinder throughout, and excessive "wire-drawing" previous to cut-off.

Exhaust Passages and Pipes.—The area of section of exhaust passages should be such that the mean velocity of steam does not exceed 6000 feet per minute, and if the distance from the cylinder to the condenser is comparatively great, a somewhat larger area is advisable. There should not be a greater difference than 1 lb. between the pressure in the cylinder and that in the condenser when exhausting.

Receiver Space. — The space between the valve of the high-pressure cylinder and that of the low pressure cylinder, into which the steam exhausts from the high-pressure cylinder, should

be from 1 to 1.5 times the capacity of the high-pressure cylinder, when the cranks are set at an angle of from 120° to 90°. When the cranks are opposite or nearly so, this space may be very much reduced. The pressure in the receiver should never exceed half the boiler pressure, and is generally much lower than this. It is usual to fit a safety valve to the receiver, loaded by weight or spring to a pressure of 20 to 30 lbs. per square inch; otherwise, owing to the large flat sides between the two cylinders, great risk of explosion would be run. This safety valve is usually of the same size and design as the cylinder escape valves. The receivers of three-crank engines need not be nearly so large, as the cranks are usually at angles of 120°; in the case of triple compound engines with the M. P. leading the H. P., a very small receiver will do.

CONNECTING RODS AND BRASSES.

The following empirical formula will be found a very useful one, and the results given by it agree very closely with good modern practice.

$$\text{Diameter of connecting-rod at middle} = \frac{\sqrt{L \times K}}{4}.$$

L is the length of the rod in inches, and—

$$K = 0.03 \times \sqrt{\text{Effective load on the piston in lbs.}}$$

Example.—To find the diameter of the con-

necting-rod, 100 inches long, for an engine having a load of 55,000 lbs.

$$K = 0.03 \times \sqrt{55,000} = 7.0.$$

$$\text{Diameter} = \frac{\sqrt{100 \times 7}}{4} = 6.6 \text{ inches.}$$

The diameter of the connecting-rod at the ends may be 0.875 of its diameter in the middle. The tapering of rods, or making them barrel-shaped, is usual in the case of those having single brasses at both ends, such as are generally fitted to trunk and return connecting-rod engines; then the diameter of the crank-pin end is 0.925 of the diameter at middle. Direct-acting engines have usually the connecting-rods tapering from the gudgeon end to the middle, and then parallel or nearly so to the crank-pin end.

Connecting-Rod Bolts.—The diameter of the bolts may be calculated by allowing the same strain per square inch as that given for piston-rod bolts. It is usual now, from practical considerations, to make the bolts of both piston and connecting-rod of the same size; the bolts therefore should be calculated from the strain on the connecting-rod. In order that the whole of the stretch shall not come on one section, as at the bottom of the last thread of an ordinary bolt, it is better to turn part of the body of connecting and piston-rod bolts to the same diameter as at the bottom of the thread, leaving it a little larger

than the diameter over the thread close to the head, and in way of any joint—that is, the bolt is made with a *plus* thread, and bearing collars where required.

Connecting-Rod Brasses. — The crank-pin brasses are more severely tried than any others about an engine, and, therefore, should be most carefully designed, and made of the very best material. Some engineers make the brasses to form the end of the rod (Fig. 29), and retained to it by bolts and a wrought-iron cap; others prefer that they shall only act as bushes or liners to the connecting-rod, sometimes fitting them into a square or octagonal recess in the rod end, and held in place by a flat cap and bolts, just as is generally done to piston-rod ends; but more generally they are fitted in duplicate halves, as shown in Fig. 30. The former plan is a very expensive one when they are of large size, on account of the great weight of brass required, and consequently are also costly to renew when worn, besides which they are very liable to get out of shape when heated, and to crack through the crowns. The latter plan avoids the use of so much brass, gives a good solid bed to the brasses, and leaves the bolts free of all strain except tension. When rods are made in this way it is usual to forge the head of the rod solid, and turn it and the cap at the same operation; the hole for the brasses is bored or slotted out

(the latter when the hole is 9 inches and upwards in diameter) roughly; the head is then slotted through or parted in the lathe so as to cut off the cap, the space left by the tool being equal to twice the difference in thickness of the brass at the crown and sides; the cap is then bolted close to the rod, and the hole bored out to the diameter of the brasses measured across the rod. The brasses are kept from turning by a brass distance-piece secured between the cap and rod and projecting between the brasses, and in the case of large brasses a short feather is fitted close to each flange in the crown. All brasses have a tendency to close on the pin or journal after having been hot, because the inner surface becomes warm first, and the metal in expanding tends to straighten the curved part; this is resisted by the other part of the brass and the bed in which it is fitted, and in consequence this inner surface gets compressed permanently, so that on cooling down it contracts, and tries then to give the brass more curvature, and so presses hard on the journal. This is the reason why some bearings will never work cool but always a trifle warm, this slight amount of heat causing the brass to expand so as to truly fit the journal.

"Magnolia" metal is better than bronze for the rubbing surface of the crank-pin brasses, and it is important that the metal shall project beyond the brass so that it alone bears on the pin.

For this purpose strips of metal should be fitted into grooves planed in the brass, and be well hammered, so as to thoroughly fill the spaces, after which it should be smoothly bored and fitted to the pin. Brasses which have not been originally designed for white metal may be fitted in this way, or by boring some shallow holes, whose diameter at the bottom is more than at the surface, casting into them buttons of white metal, and after hammering down, boring out the brass so that the white metal stands out beyond the original wearing surface.

A very good plan, but somewhat more expensive and not more efficient than the one above described, is to run the metal into recesses so cast with the brass, hammer it well in place, bore out, and then plane out the brass intervening between the white metal patches, leaving only slight ridges surrounding the latter, which prevent it from being spread out.

Caps of Connecting-rod Brasses.—The width

hould be a little

he end; its thick-

gth)=0.6 × diam-

ldle = 0.8 × diam-

pitch of bolts

= diameter of bolts

caps are generally

made straight and of thickness given by the first rule.

Gudgeon End Rod.—Direct-acting engines have usually the connecting-rod formed with a fork at the gudgeon end (Fig. 29), into which the gudgeon is shrunk.

The diameter of the eye = diameter of gudgeon + 0.9 × diameter of connecting-rod at end.

The thickness of metal around gudgeon= 0.55 × connecting-rod at end.

Width of jaw = 1.1 × diameter of rod at end.

Thickness of jaw = 0.45 × diameter of rod at end.

MAIN BEARINGS.

The bearings in which the shaft journals run should approximate, as far as possible, to a hole through a solid support. If it were possible, a hole with a bush of suitable metal in it would form the best possible bearing for a shaft; but since the bearing, however well designed and made, will, in course of time, wear somewhat, it becomes a necessity that there shall be some means of adjusting the brasses, so as to prevent the shaft from having side movement when they are worn. In the case of the vertical engine, the weight of the shaft, and the pressure from the piston, act very nearly in the same direction, so that the wear is only vertically above and below the shaft; consequently the adjustment is necessary only in a vertical direction. The greatest

strains on the bearings, however, are during the first half of the stroke, and consequently the position of mean pressure on the journals is not exactly vertical; this is also somewhat modified on the upstroke by the tendency of the shaft to roll on the surface of the brasses, and on the downstroke it is aggravated from the same cause. In fitting the brasses for a vertical engine, this should be borne in mind, and every allowance made for taking the wear due to these causes. It is of the utmost importance for the good working and endurance of a crank-shaft, that the bearings are rigid in themselves, and that the framework containing them shall be rigid enough to sustain them perfectly in line one with another. Crank-shafts are more severely tried by the giving or springing of the bearings than any other cause, and they are oftener broken from want of rigidity in the bed-plate and seatings, than from the normal strains from the pistons, so that a shaft may be of ample size to bear the twisting and bending strains if properly supported in its bearings, and yet give way after a few weeks' work in a weak ship.

The brasses are usually formed as shown in Fig. 31, and carefully bedded into the recesses provided for them in the foundation. At one time it was usual to design them with projecting facings, called chipping strips, to avoid the labor of chipping and filing the whole of the

surface; this was, however, found to be highly objectionable as engines increased in size; and with the increase of boiler pressure, and consequent increased percussive action due to the high initial pressure, such an effect was produced on these strips, and the cast-iron surface on which they were borne, that engineers have gradually abandoned the practice; the planing machines also have rendered such a device unnecessary, as it is nearly as cheap to fit brasses so as to bear over the whole surface as to do so only on strips. The square bottom brass is objectionable on two grounds; one being that it is impossible to remove it in most engines without lifting the shaft, and the other that when it becomes hot it is invariably distorted, from its variation in thickness of metal, with the result that it is broken through the middle longitudinally.

The first of these evils is avoided by making the bottom brass round and of even thickness, so that it can be got out when relieved of the weight of the shaft, by being moved around until it is on the top of the journal. The second evil is also partly avoided by making it of an even thickness; but this form of brass is often found cracked, and is liable to heat from its want of stiffness. Both these brasses, when first heated by abnormal friction, tend to expand along the surface in contact with the shaft; this would open the brass, and make the bore of

larger diameter, if not prevented by the cooler part near the cast iron, and by the bed-plate itself. If the brass has become hot quickly and

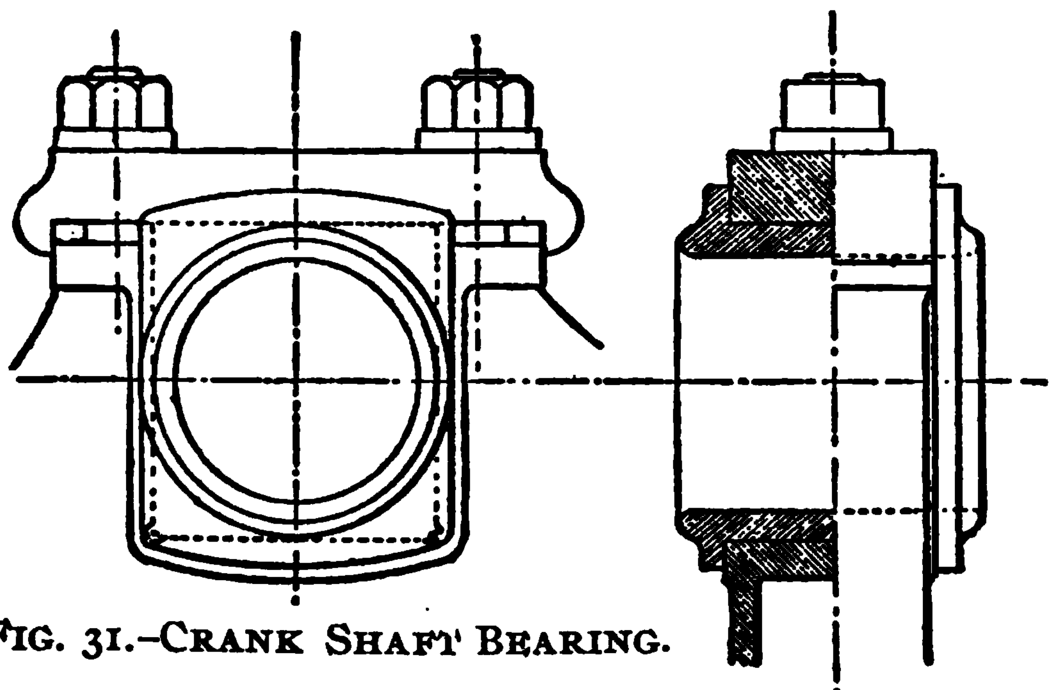

FIG. 31.—CRANK SHAFT BEARING.

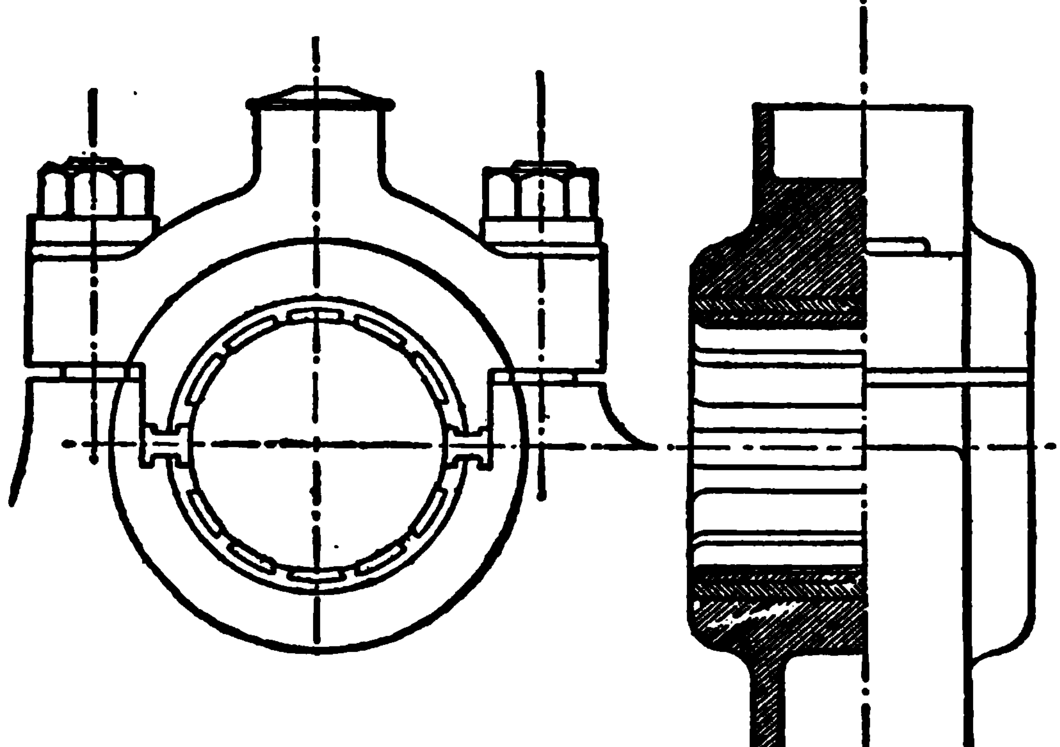

FIG. 32.—IMPROVED FORM OF CRANK-SHAFT BEARING.

excessively, the resistance to expansion produces permanent set on the layers of metal near the journal, so that on cooling the brass closes and

tends to grip the shaft; it will then set up sufficient friction to heat again, and expand sufficiently to ease itself from the shaft, and so long as that temperature is maintained the shaft runs easily in the bearing. This is why some bearings always are a trifle warm, and will not work cool. A continuance of heating and cooling will set up a mechanical action at the middle of the brass, which must end in rupturing it, just as a piece of sheet metal is broken by continually bending backwards and forwards about a certain line.

This action of the brass can be prevented by securing it to the bed-plate, along its two longitudinal edges, as shown in Fig. 32 by an **H** shaped strip, which holds both top and bottom brasses, so that they cannot move in their beds. This method is a very simple one, and has been most successful in engines of all sizes.

It is also essential that the bearing, to be efficient, should be rigid throughout its whole length; this is not the case when the brasses have long overhanging ends, which afford little or no support to the shaft. To this end it is better, when possible, to extend the bed for the brasses, so as to support them over the whole of their length, as shown in Fig. 32.

Caps for Main Bearings are very generally made of wrought iron, but as stiffness is as necessary as strength, cast iron may be used with

advantage in their construction. A wrought-iron cap, which may be amply strong, is often far from stiff enough, while a cast-iron cap, which is stiff enough for good working, is generally amply strong.

MAIN-BEARING BOLTS.

Each cap is usually held by two bolts, but very large bearings have four bolts, two on each side, so as to avoid large bolts and heavy nuts, and to distribute the strain over the cap. When everything is in good order and properly adjusted, the strain from the piston should be equally divided between the bolts; but since, from a very slight difference in setting of the nuts, the strain may come on three, and sometimes even on two bolts only, due allowance must be made for this.

BRASSES.

Brasses, so called because generally made of brass. They should be made of a metal which will withstand wear without wearing the shaft journals, and whose surface is such that the shaft runs on it with a minimum amount of friction. The metal must also possess sufficient strength, so as not to fracture under the percussive strains of the piston, and be free from brittleness, so as not to crack when quickly cooled. Good gun-metal or bronze possesses *all*

the qualities essential for brasses, but there are other metals which have certain of these qualities in a higher degree without having them all. Cast iron is harder than ordinary bearing bronze, and when once worn to a smooth surface gives equally good results; but it is liable to fracture from continued shocks and when cooled suddenly. White metals offer least resistance, or produce least friction, but most of them are too soft to be used alone. Of the patent bronzes there are few which are suitable for heavy bearings, and none of them have so far been shown to be much superior to good gun-metal.

When a bearing is of ample size, properly designed and constructed, and well looked after, it may be of almost any kind of metal. If the bearing surface is limited, there is a great difference in the behavior of different metals; and if badly designed and constructed, even the best metal will give trouble; but if not properly looked after by the engineer, the best metal and the most careful design are of no avail.

Magnolia metal has so far given the best results as a bearing surface, and there is every reason for this, inasmuch as it is too soft to cause abrasion of the shaft, and if its own surface is injured it will not form into fine sand, and grind both the surfaces, as all the bronzes do more or less. When it is used it is highly important that the shaft shall bear wholly on it, and not

partly on it and partly on the metal containing it, and also that efficient courses for the distribution of the lubricant are provided.

CRANK SHAFTS.

$$\text{Diameter of shaft} = \sqrt[3]{\frac{\text{I. H. P.}}{\text{R}}} \times 35.7.$$

But as shafts must be strong enough to resist the *maximum* twisting strain, it is necessary always to base calculations on it instead of on the mean twisting moment.

LINE SHAFTING.

$$\text{Diameter of the line shafts} = \sqrt[3]{\frac{\text{I.H.P.}}{\text{R}}} \times \text{F}.$$

Compound engines, cranks at right angles—

Boiler pressure 70 lbs.,	rate of expansion	6 to 7,	F = 70.
" 80	"	7 to 8,	F = 72.
" 90	"	8 to 9,	F = 75.

Triple expansion, three-cranks at 120°—.

Boiler pressure 150 lbs.,	rate of expansion	10 to 12,	F = 62.
" 160	"	11 to 13,	F = 64.
" 170	"	12 to 15,	F = 67.

Expansive engines, cranks at right angles, and the rate of expansion 5, boiler pressure, 60 lbs., F = 90.

Single crank compound engines, pressure 80 lbs., F = 96.

SCREW PROPELLERS.

The area of blades given by the following rule is such as is generally found to give good results, and may be used by those who have no good experience to guide them:—

$$\text{Total area of screw-blades} = K\sqrt{\frac{\text{I. H. P.}}{\text{revolutions}}}.$$

The value of K,	for	four-bladed	propellers,	is	15.
"	"	three-	"		13.
"	"	two-	"		10.

THRUST BEARING.

D = diameter of thrust collar.

d = diameter of shaft.

In practice the thickness of each collar $= 0.4 (D-d)$.

(1) Space between the collars, if rings are of solid brass $= 0.4 (D-d)$.

(2) Space between the collars, if rings are of cast iron faced with brass or white metal $= 0.75 (D-d)$.

(3.) Space between the collars, if rings are of hollow brass for water to circulate through $= D-d$.

The number of collars depends very much on the size of the engine and the prejudice of the designer. If there are many collars, they are of necessity somewhat small, and although the *chances* are in favor of the majority of them acting efficiently, allowance must be made for the

contingency of the whole thrust coming only on one of them, and the larger the number of collars, the less able is each one separately to resist the whole thrust. The chief objection to a few collars is that they are of necessity of comparatively large diameter, and have, therefore, a higher speed of rubbing surface; there is also the consideration of cost of forging against large collars.

When there are a few large collars, a better de-

FIG. 33.—COMMON THRUST-BLOCK.

sign of thrust-block is possible, and the rings can be made adjusiable without removal.

The number of collars should vary with the size of the shaft, and a very good rule is that there should be one collar for shafts up to 6 inches diameter, and then an additional collar for every 2 inches of diameter beyond this.

The common plan of thrust-block with many collars is shown in Fig. 33, and a modification of the same is made by having the rings in one

casting. These plans are cheap, and do very well for small engines. So long as no heating is allowed to take place in the bearing, it will work very well; but when once it gets out of order it is difficult to deal with, and impossible to adjust at sea. It suffers from being enclosed, and from the rings lacking means of independent adjustment. Fig. 34 shows a plan of thrust-block which is most suitable when there are a few large collars. Here the thrust is taken by

FIG. 34.

horse-shoe shaped pieces of metal faced with brass or white metal, and fitted sometimes carefully into recesses on either side of the main block. When faced with brass, each may be adjusted very simply by putting thin tin liners behind the facings, which are hung on steady pins.

Figs. 34 and 35 are an elaboration of the form of block. Here the horse-shoes fit over two screwed bars, one on either side of the block; nuts are fitted to these bars, so that each collar may be adjusted by its own nuts, or the whole of them by the nuts at the end.

Both these plans are most successful in practice, in great measure due to the fact that the collars are open and exposed at the top, so as to be easily lubricated and cooled by the air, and to their running in oil, or in a mixture of oil and soapy water contained in the trough below them.

It is most important that a bearing be placed close to the thrust, so that the shaft cannot vibrate and cause uneven pressure over the surface of the collars. The function of the thrust

FIG. 35.

bearing is to take only end pressure. This is particularly the case when designed with horseshoe rings.

THE CONDENSER.

The function of the condenser is to so cool down the exhaust steam as to reduce its pressure to a minimum, and in doing so the steam is converted into water.

Condensing engines require from 20 to 30 gallons of water to condense the steam represented

by every gallon of water evaporated—approximately for most engines, we say, from 1 to 1½ gallons per minute per I. H. P. Jet condensers do not require quite as much water for condensing as surface condensers.

Surface condensers should have about 2 square feet of tube (cooling) surface per horse-power of steam engine. It is absolutely necessary to place air-pumps below condensers to get satisfactory results.

In general practice the following holds good when the temperature of sea-water is about 60°:

Terminal pressure,	30 lbs.	absolute,	3 square	feet	per I.H.P.
"	20	"	2.50	"	"
"	15	"	2.25	"	"
"	12½	"	2.00	"	"
"	10	"	1.80	"	"
"	8	"	1.60	"	"
"	6	"	1.50	"	"

For ships whose station is in the tropics the allowance should be increased by 20 per cent., and for ships which occasionally visit the tropics 10 per cent. increase will give satisfactory results. If a ship is constantly employed in cold climates, 10 per cent. less suffices.

AIR PUMPS.

The function of this pump in all condensers is to abstract the water condensed, and the air which was originally contained in the water

when it entered the boiler; and in the case of jet condensers, it pumps out in addition the water of condensation and the air which it contained.

The Single-Acting Vertical Air-Pump, having valves in the bucket as well as foot and delivery valves, is by far the most efficient, and, when possible, is generally chosen. If the rod of such a pump is enlarged or the bucket has a trunk to surround the rod, which is attached to a joint in its centre, it is to a certain extent double-acting, since on the upstroke it fills the chamber in which it works, on the downstroke it displaces, and consequently discharges a volume equal to the volume of the trunk or rod, and on the upstroke discharges the remainder. If the sectional area of the rod or trunk is half that of the bucket, the discharges are equal, and the pump is virtually a double-acting one.

The following table gives the ratio of capacity of cylinder or cylinders to that of the air-pump; in the case of the compound engine, the low-pressure cylinder capacity only is taken.

Description of Pump.	Description of Engine.	Ratio.
Single-acting vertical.	Jet-condensing, expansion 1½ to 2	6 to 8
" "	Surface " " "	8 to 10
" "	Jet " " 3 to 5	10 to 12
" "	Surface " " "	12 to 15
" "	" " compound	15 to 18
Double-acting horizontal.	Jet-condensing, expansion 1½ to 2	10 to 13
" "	Surface " " "	13 to 16
" "	Jet " " 3 to 5	16 to 19
" "	Surface " " "	19 to 24
" "	" " compound	24 to 28

PUMP BUCKETS.

The following rules give the dimensions of an ordinary pump-bucket, of which Fig. 36 is an example:—

$$x = 0.3 \times \sqrt[4]{D} + 0.15 \text{ inch.}$$

D is the diameter of the pump in inches.

The thickness of the disc when solid	$= 1.0 \times x$.
" " " perforated	$= 1.7 \times x$.
" " flanges at edge	$= 1.1 \times x$.
" " metal around rod end	$= 1.5 \times x$.
" " " the rim	$= 1.0 \times x$.
" " packing	$= 1.1 \times x$.
" " ribs	$= 0.8 \times x$.
The breadth of the packing	$= 4.0 \times x$.
The depth of bucket at the middle	$= 6.0 \times x$.

The number of ribs, one for each 4 inches of diameter.

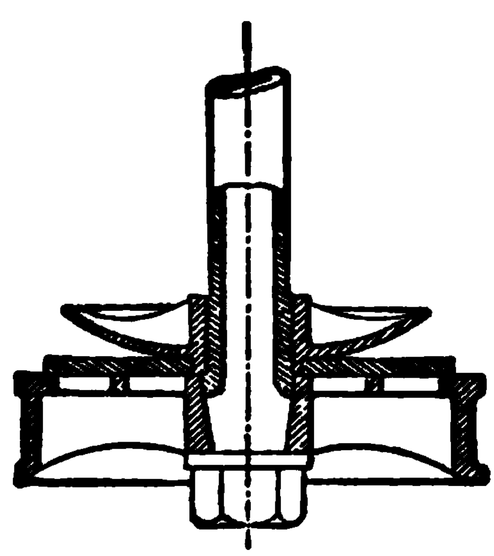

FIG. 36.

The brass liner is usually from ½ inch to ⅞ inch diameter, or its thickness $= 1.1\,x - 0.2$ inch.

The size of the circulating pump is, to a large extent, dependent on the same conditions that determine the size of the air-pump, and may,

therefore, bear a constant relation to the size of the air-pump; and since the size of the air-pump is often determined by the size of the cylinders, that of the circulating pump may be found in a similar manner. When the air-pump is *single-acting* the capacity of the *single-acting* circulating pump should be 0.6 of that of the air-pump, and when the circulating pump is *double-acting*, 0.31.

When the air-pump is *double-acting*, the capacity of the *double-acting* circulating pump should be 0.52 of that of the air-pump, the double-acting circulating pump being more efficient than the double-acting air-pump.

The following table gives the ratio of capacity of cylinder or cylinders to that of the circulating pump.

Description of Pump.	Description of Engine.	Ratio.
Single-acting.	Expansive 1½ to 2 times.	13 to 16
" "	" 3 to 5 "	20 to 25
" "	Compound	25 to 30
Double-acting.	Expansive 1½ to 2 times.	25 to 30
" "	" 3 to 5 "	36 to 46
" "	Compound	46 to 56

WHEELER'S IMPROVED SURFACE CONDENSER.

Figs. 37 and 38 show the combination of the Wheeler Surface Condenser with independent

Fig. 37.—Wheeler's Improved Surface Condense

DENSER. COMBINED WITH INDEPENDENT AIR AND CIRCULATING PUMP.

To Face Page 106.

system of air and circulating pumps. The condenser is mounted on the latter, thus saving floor space, which is of considerable importance in engine rooms that are crowded with machinery, such as are found on many steam vessels, and in numerous factories located in crowded towns and cities.

In this design, the condenser is firmly bolted down on the pumps, the latter serving as a foundation, thereby saving expense of piping between the pumps and the condenser. As shown by the sectional view, the water pump delivers the circulating water through the nozzle *C* directly into the lower sections of tubes in the condenser. From thence the water flows by the passageway *E* into the chamber *H*, passing into the upper group of tubes, and finally discharging through the outlet nozzle *D*, as shown by the arrows.

The exhaust steam entering at the nozzle A comes in contact with the scattering plate *O*, thereby distributing uniformly over the cooling surface of the tubes. The water of condensation gravitates to the bottom of the condenser, and flows directly to the air pump by the annular passage *B*, the air pump discharging the water through the outlet nozzle at the side of the cylinder. Both air and circulating pumps are operated by a direct-acting steam cylinder in the usual way.

As shown by the sectional view, the arrange-

ment is one providing very thoroughly for the expansion and contraction of the tubes, and this without the use of tube packings of wood, paper and similar materials. In fact, there are no ferrules, washers or packings of any kind employed.

The tubes are seamless drawn brass, carefully tinned inside and outside. The tubes are arranged in pairs, the smaller tube inside the larger. The latter is thickened at one end, on which a substantial deep thread is chased. This end of the tube is screwed into a head of brass, and on the other end of the tube is screwed a cap, as shown. One end of the small tube is also drawn thick, and a thread chased on it. This tube is also screwed into a head of brass. The tubes can be easily taken out and thoroughly cleaned, as their form and the manner of fastening permit of this being readily done.

The tightly screwed fastenings for tubes also permit the use of circulating water under pressure, which is often the case where the supply is from a head of considerable height; in other words, there are no tube packings of any kind to be forced out or loosened by pressure of the circulating water.

The circulation of water is very thorough, and consequently the smallest amount of cooling water is required. This feature gives a marked saving in the power necessary for circulating pump.

FIG. 38.—WHEELER'S IMPROVED SU

) SURFACE CONDENSER. SECTIONAL VIEW.

TO FACE PAGE 1[illegible].

In a test recently made by experts with a Wheeler surface condenser, the temperature of injection water was 56 1-2 degrees, temperature of discharge 98 degrees, temperature of hot well 138 degrees, and vacuum 24 1-2 inches, 101.8 pounds of steam was condensed per hour per square foot of condensing surface.

In another test the condenser was worked as a simple surface condenser without vacuum, and with injection 78 1-2 degrees, discharge 139 degrees, hot well 201 degrees, 204.2 pounds of steam per hour per square foot of condensing surface was condensed.

The condensers are made with and without pumps attached, and with bodies of both circular and rectangular shape. The larger size are usually of the latter design.

INDEPENDENT AIR AND CIRCLUATING PUMPING ENGINE.

The United States battle ship *Maine* is to have an entire outfit of steam pumps of the "Blake" type, including the independent air and circulating pumps for the marine engines (see Fig. 39).

The general design of this combined air and circulating pumping engine is similar to that now in use on the United States cruisers *Chicago* and *Dolphin* (which were also furnished by the same company), with the exception of the steam end of the machine, which is arranged on the

compound system in order to economize steam. There will be two of these pumping engines for the battle ship *Maine* (one for each main engine), and the dimensions of each are as follows: High pressure steam cylinder, 12 inches in diameter; low pressure steam cylinder, 24 inches in diameter; air pumps, each 30 inches in diameter; circulating water pumps, 31 inches in diameter; the length of stroke is less than two feet, owing to the limited space that could be given for floor space.

As will be seen by the Figure (39), the air pumps are of the single acting vertical type, worked by a beam, which is operated by means of the rock shaft from the direct-acting pumping engine. The water cylinder is of the ordinary double acting horizontal type, but has an unusually large amount of valve area, so as to admit of high piston speed, the object being to reduce weight and space to a minimum, and yet secure large pumping capacity. The high and low pressure steam cylinders are operated by the "Blake" valve gear, which is adjustable while the pump is in operation, and is perfectly positive at any and all speeds. The adaptability of the direct acting engine to the varying and uncertain load of an air pump makes this system very satisfactory. The pumps of the U. S. S. *Chicago*, *Dolphin*, *Boston* and *Atlanta* (also those of the same type on the U. S. S. *Concord*) have given entire satisfaction.

FIG. 39.—INDEPENDENT AIR

r Air and Circulating Pumping Engine.

Opposite Page 1[illegible].

The air buckets of this machine are of special construction, which avoids any possibility of air pockets, and at the same time drains the channel way very effectively of water, so as to assist the free flow of vapors. The air and pump cylinders are made entirely of gun metal composition, to comply with the government specifications. This secures great durability and light weight. The water cylinder is provided with a by-pass valve (not shown), so that the amount of circulating water can be regulated according to its temperature, which, of course, varies with the seasons of the year and other conditions.

The capacity of these independent air and circulating pumping engines is for 4,000 I. H. P. each, which is the maximum estimated power of each of the triple expansion engines of the battle ship *Maine.*

FEED PUMPS.

The duty of the feed pump is to supply the boiler with sufficient water to meet its wants. It is supplied from the hot-well, so that when there is a surface condenser its supply is fresh water, and the amount, under ordinary circumstances, is the same as that evaporated in the boiler; but owing to leakage and waste by blowing the steam whistle, or using an auxiliary engine which exhausts into the air, the quantity

of water condensed is not always sufficient to make up for that evaporated. It is found necessary also occasionally to blow some of the water out of the boiler, to get rid of scum floating on the surface of the water, and this waste must be made good by a supply from the sea. The water from a jet condenser is very nearly as salt as sea water, and this, on being evaporated in the boiler, leaves behind the salt, etc., so that unless some precautions were taken the boiler would in time become filled with solid matter. To prevent such a large deposit of salt, etc., in the boiler, it is customary to blow out some of the very dense water from the boiler at fixed intervals, and as the blow-off cock is situated near the bottom of the boiler, a considerable amount of solid matter is thus got rid of; the quantity of water blown out is made up by an extra supply from the hot-well. Now, since the surface condenser may leak, or an accident may happen, whereby jet condensation has to be resorted to, the pumps of engines fitted with surface condensers must be sufficiently large to do duty under such circumstances.

Since a surface condenser supplies pure water for feeding the boiler, and there is not the same need for blowing off the boilers, the feed-pumps may be very much smaller than when jet condensation is practised, and are generally of such a size that *each* is capable of delivering three times

the *net feed-water;* the pumps, when both are working, can then deliver six times the net feed, which is sufficient to satisfy the demands should jet condensation become necessary.

The following empirical formula will give such sizes as will be found in practice:

$$\text{Capacity of each feed-pump} = \frac{\text{capacity of cylinder}}{C}.$$

The following are the values of C when the engine is surface-condensing:

Terminal pressure under	25 lbs.	C = 220.
" "	20 lbs.	C = 250.
" "	15 lbs.	C = 320.
" "	$12\frac{1}{2}$ lbs.	C = 380.
" "	10 lbs.	C = 440.
Compound engines generally (taking L. P. cylinder only)		C = 400.

The net feed-water in cubic feet per stroke is approximately

= area of piston in inches × stroke in feet × absolute pressure at release ÷ 3,125,000; or,

Net feed-water in pounds per stroke approximately

= area of piston in inches × stroke in feet × absolute pressure at release ÷ 50,000.

FEED PIPES.

Feed pipes leading to and from the feed pumps should be such that the velocity of flow does not exceed 500 feet per minute, and small pumps

should have larger pipes in proportion, so that the flow through them does not exceed 400 feet. Since the amount of water actually flowing through the pipes is generally very considerably less than the pumps are capable of discharging, the velocity is seldom more than one-half the above allowance; but, as the pumps do occasionally deliver their full amount, the pipes must be large enough for that purpose.

If d is the diameter of the feed pump plunger, s its mean velocity in feet per minute, then the diameter of feed pipe $= \frac{d}{20}\sqrt{s}$ for small pumps, and diameter of feed pipe $= \frac{d}{23}\sqrt{s}$ for large pumps. *Example.*—To find the diameter of the feed pipes for a pump whose diameter is 6 inches and the length of stroke 2 feet, worked from the lever of an engine making 60 revolutions per minute. Here $s = 2 \times 60 \times 2 = 240$ feet. Diameter of pipe $= \frac{6}{23}\sqrt{240} = 4$ inches. If there are two pumps which deliver alternately, the pipes will be the same size throughout; but if the two pumps may deliver at the same time, the pipe beyond the junction of the two from the pumps must be nearly double the sectional area of one.

As the resistance of pipes is due greatly to friction at the surface, and will consequently vary as the diameter, while the area of the section varies as the square of the diameter, the re-

sistance in the single pipe will be considerably less than the combined resistance in the two, and for this reason its area may be less. In practice this area may be 0.8 of the combined area of the two. Hence, when there are two pumps delivering together, diameter of main pipe = 1.265 × diameter of branches. If there were two pumps, as in the last example, delivering together, the diameter of the main pipe would be $4 \times 1.265 = 5.06$ inches. The rating of so many horse-power of a boiler does not give us the required data for determining the size of pump and feed pipes, because some boiler makers allow 10 square feet of heating surface per horse-power, some 12, and others 15 square feet. We must know the number of pounds of steam used, or the number of pounds of water the boiler is capable of evaporating in a given time, and from this data determine the size of pump required, and then find the size of feed pipes by the foregoing rules.

Sometimes the velocity of the water in the suction pipes is slower than that in the discharge pipes, therefore the discharge pipes in such cases are made smaller than the suction pipes.

CHAPTER XV.

UNITED STATES GOVERNMENT GENERAL RULES AND REGULATIONS FOR STEAM (MARINE) BOILERS.

1. EVERY iron or steel plate intended for the construction of boilers to be used on steam-vessels shall be stamped by the manufacturer in the following manner: At the diagonal corners, at a distance of about four inches from the edges, and at or near the centre of the plate, with the name of the manufacturer, the place where manufactured, and the number of pounds tensile strain it will bear to the sectional square inch.

2. Whenever inspectors shall find a plate of iron or steel with stamps differing as to the tensile strength of the material, they shall rate the tensile strength of the same in accordance with the lowest stamp found thereon.

3. To ascertain the tensile strength of plates, a piece shall be taken from each sheet to be tested, the area of which shall equal one-quarter of one square inch on all plate $\frac{5}{16}$ inch thick and under; and all plate over $\frac{5}{16}$ inch thick the area shall equal the square of its thickness; and the force at which the piece can be parted in the

direction of the fiber or grain, represented in pounds avoirdupois—the former multiplied by four, the latter in proportion to the ratio of its area—shall be deemed the tensile strength per square inch of the plate from which the sample was taken; and should the tensile strength ascertained by the test equal that marked on the plates from which the test-pieces were taken, the plates must be allowed to be used in the construction of marine boilers: *Provided always*, That the plates possess homogeneousness, toughness, and ability to withstand the effect of repeated heating and cooling; but should these tests prove the plates to be overstamped, the lots from which the test-plates were taken must be rejected as failing to have the strength stamped thereon. But nothing herein shall be so construed as to prevent the manufacturers from restamping such plates at the lowest tensile strength indicated by the samples, provided such restamping is done previous to the use of the plates in the manufacture of marine boilers.

4. Local inspectors are required to make and send to the supervising inspectors, with every sample of iron or steel to be tested, their *certificate* that the sample sent for testing was cut from the plate or plates to be used in the boiler designated. And the manufacturer of any boiler to be used for marine purposes shall furnish the inspectors an affidavit in the following form, sub-

scribed to either by himself or authorized agent having superintendence of the construction of such boilers:

AFFIDAVIT OF MANUFACTURER OF MARINE STEAM-BOILERS.

COUNTY OF ——,

State of ——, *ss:*

On this —— day of ——, A. D. 189—, personally appeared before me. a notary public in and for the county of —— and State of ————, Mr. —— ——, who, being duly sworn, deposes and says that he is —— of —— ——, boiler manufacturer, and that the accompanying samples of ——, manufactured by —— ——, of ——, were cut from plates stamped —— T. S., which are to be used in the construction of a marine boiler for ——, and no plate of less tensile strength or quality than herein specified will be used in the construction of said boiler, the dimensions of which will be ——, and of the style known as ——, to be used upon the steamer ——.

—— ——,

——.

Sworn and subscribed to before me this —— day of ——, 189—.

—— ——,

Notary Public.

6. To ascertain the ductility and other lawful qualities, iron of 45,000 pounds tensile strength, and under, shall show a contraction of area of 15 per cent., and each additional 1,000 pounds tensile strength shall show one per cent. additional contraction of area, up to and including 55,000 T. S. Iron of 55,000 T. S. and upwards, showing twenty-five per cent. reduction of area, shall be deemed to have the lawful ductility. All steel plate of one-half inch thickness and under

shall show a contraction of area of not less than fifty (50) per cent. Steel plate over one-half inch in thickness shall show a reduction of not less than forty-five (45) per cent. *Provided, however,* That steel plate required for repairs to boilers built previous to April 1, 1886, may be used for such repairs when showing a contraction of area of not less than forty (40) per cent.

7. In the following table will be found the width—*expressed in hundredths of an inch*—that will equal, near enough for practical purposes, one-quarter of one square inch of section of the various thicknesses of boiler plates:

$\frac{3}{16}''$ × 133	.26 × 96	.35 × 71
.21 × 119	.29 × 86	$\frac{3}{8}''$ × 67
.23 × 109	$\frac{5}{16}''$ × 80	$\frac{7}{16}''$ × 57
$\frac{1}{4}''$ × 100	.33 × 76	$\frac{1}{2}''$ × 50

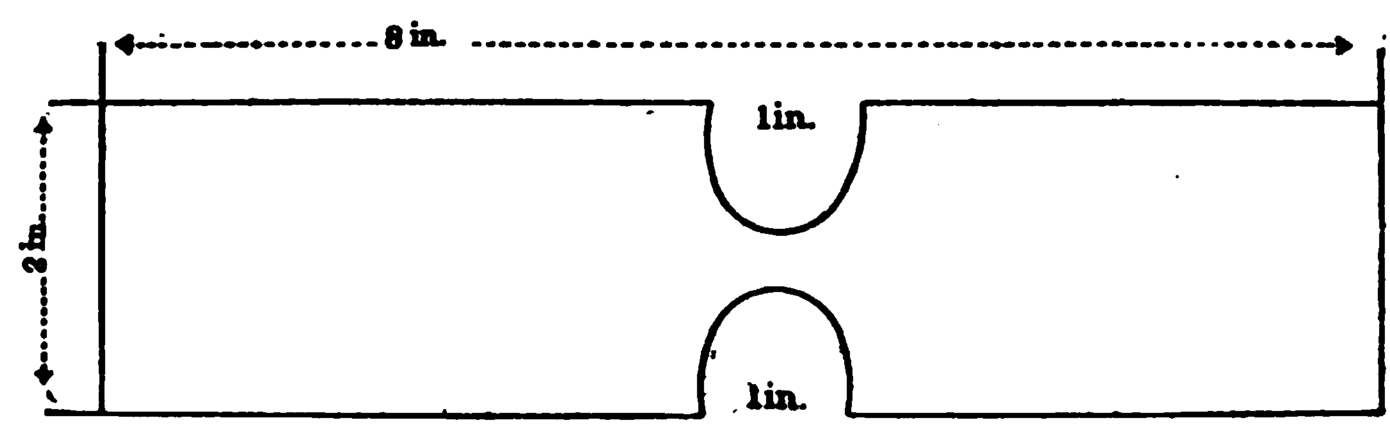

FIG. 40.

8. The gauge to be employed by inspectors to determine the thickness of boiler-plates, and the widths in the table, will be any standard American gauge furnished by the Treasury Department.

9. All samples intended to be tested on the Riehlé, Fairbanks, or other reliable testing-machine, must be prepared in form according to Fig. 40, viz., eight inches in length, two inches in width, cut out at their centers in the manner indicated:

10. All tests made of boiler material must be recorded upon a table of the following form:

TENSILE TESTS OF SAMPLES OF MATERIAL INTENDED TO BE EMPLOYED IN THE CONSTRUCTION OF BOILERS OF STEAM-VESSELS MADE ON ——— TESTING-MACHINE.

ate when tests were made.	rom whom samples were obtained, and by whom tested.	laterial, iron or steel.	tamp or label on samples, which must be the same as stamps on the material from which they are taken.	hickness of samples, expressed in hundredths of an inch.	Vidth of samples, expressed in hundredths of an inch.	train at which each sample parted.	train per square inch of section.	Reduced thickness.	Reduced width.	Contraction of area—per cent.	Remarks.

(1) PRESSURE ALLOWABLE ON BOILERS OF VARIOUS DIMENSIONS BUILT PRIOR TO FEBRUARY 28, 1872.

1. Boilers built prior to February 28, 1872, shall be deemed to have a tensile strength of 50,000 pounds to the sectional square inch, whether stamped or not, and shall be tested under the rule prescribed for boilers inspected under the provisions of section 36 of the act relating to boilers built after the 28th of February, 1872.

* * * * * * *

3. The pressure for any dimension of boilers must be ascertained by the following rule, viz:

Multiply one sixth ($\frac{1}{6}$) of the lowest tensile strength found stamped on any plate in the cylindrical shell by the thickness—expressed in inches or parts of an inch—of the thinest plate in the same cylindrical shell, and divide by the radius or half diameter—also expressed in inches—and the sum will be the pressure allowable per square inch of surface for single riveting, to which add twenty per centum for double-riveting.

4. The hydrostatic pressure applied must be in proportion of one hundred and fifty pounds to the square inch to one hundred pounds to the square inch of the steam pressure allowed.

5. Where flat surfaces exist, the inspector

must satisfy himself that the bracing and all other parts of the boiler are of equal strength with the shell, and he must also, after applying the hydrostatic test, thoroughly examine every part of the boiler.

6. No braces or stays hereafter employed in the construction of boilers shall be allowed a greater strain than six thousand (6,000) pounds per square inch of section, and no screw stay bolt shall be allowed to be used in the construction of marine boilers in which salt water is used to generate steam. But such screw stay bolts may be used in staying the fire-boxes and furnaces of such boilers, and not elsewhere, when fresh water is used for generating steam in said boilers.

7. Plates of iron or steel, used in the construction of boilers, extending beyond the cylindrical shell to the front of the boiler over the furnaces, shall extend at least 12 inches below the centre of the shell, and shall not be of less tensile strength or thickness than the adjoining sheets in the cylindrical portions of the shell.

RIVETED FLUES.

8. Flues having a diameter of from 6 inches to 7 inches shall have a thickness of material of not less than .18 of an inch, and if, in the opinion of the inspectors, it is deemed safe, be allowed a steam pressure, according to the thickness of material, as follows:

Thickness of material, .18, .19, .20, .21.

Pressure in pounds, 189, 194, 199, 204.

Flues having a diameter of over 7 inches, and not over 8 inches, shall have a thickness of material of not less than .20 of an inch, and if, in the opinion of the inspectors, it is deemed safe, may be allowed a steam pressure, according to the thickness of the material, as follows:

Thickness of material, .20, .21, .22, .23, .24.

Pressure in pounds, 184, 189, 194, 199, 204.

Flues having a diameter over 8 inches, and not over nine inches, shall have a thickness of material of not less than .21 of an inch, and if, in the opinion of the inspectors, it is deemed safe, may be allowed a steam pressure, according to the thickness of material, as follows:

Thickness of material, .21, .22, .23, .24, .25, .26.

Pressure in pounds, 179, 184, 189, 194, 199, 204.

Flues having a diameter over 9 inches, and not over 10 inches, shall have a thickness of material of not less than .21 of an inch, and if, in the opinion of the inspectors, it is deemed safe, may be allowed a steam pressure, according to the thickness of material, as follows:

Thickness of material, .21, .22, .23, .24, .25, .26, .27.

Pressure in pounds, 174, 179, 184, 189, 194, 199, 204.

Flues having a diameter over 10 inches, and not over 11 inches, shall have a thickness of material of not less than .22 of an inch, and if, in the opinion of the inspectors, it shall be deemed safe, may be allowed a steam pressure, according to the thickness of material, as follows:

Thickness of material, .22, .23, .24, 25, .26, .27, .28, .29.

Pressure in pounds, 169, 174, 179, 184, 189, 194, 199, 204.

Flues having a diameter over 11 inches, and not over 12 inches, shall have a thickness of material of not less than .22 of an inch, and if, in the opinion of the inspectors, it is deemed safe, may be allowed a steam pressure, according to the thickness of material, as follows:

Thickness of material, .22, .23, .24, .25, .26, .27, .28, .29, .30.

Pressure in pounds, 164, 169, 174, 179, 184, 189, 194, 199, 204.

Flues having a diameter of over 12 inches, and not over 13 inches, shall have a thickness of material of not less than .23 of an inch, and if, in the opinion of the inspectors, it shall be deemed safe, may be allowed a steam pressure, according to the thickness of material, as follows:

Thickness of material, .23, .24, .25, .26, .27, .28, .29, .30, .31, .32.

Pressure in pounds, 159, 164, 169, 174, 179, 184, 189, 194, 199, 204.

Flues having a diameter over 13 inches, and not over 14 inches, shall have a thickness of material of not less than .24 of an inch, and if, in the opinion of the inspectors, it is deemed safe, may be allowed a steam pressure, according to the thickness of material, as follows:

Thickness of material, .24, .25, .26, .27, .28, .29, .30, .31, .32, .33, .34.

Pressure in pounds, 154, 159, 164, 169, 174, 179, 184, 189, 194, 199, 204.

Flues having a diameter over 14 inches, and not over 15 inches, shall have a thickness of material of not less than .25 of an inch, and if, in the opinion of the inspectors, it is deemed safe, may be allowed a steam pressure, according to the thickness of material, as follows:

Thickness of material, .25, .26, .27, .28, .29, .30, .31, .32, .33, .34, .35.

Pressure in pounds, 147, 152, 157, 162, 167, 172, 177, 182, 187, 192, 197.

Flues having a diameter over 15 inches, and not over 16 inches, shall have a thickness of material of not less than .27 of an inch, and if, in the opinion of the inspectors, it is deemed safe, may be allowed a steam pressure, according to the thickness of material, as follows:

Thickness of material, .26, .27, .28, .29, .30, .31, .32, .33, .34, .35, .36.

Pressure in pounds, 140, 145, 150, 155, 160, 165, 170, 175, 180, 185, 190.

LAP-WELDED FLUES.

9. Lap-welded flues, 16 inches and not less than 7 inches in diameter:

To determine the pressure per square inch allowable on lap-welded flues 18 feet and less in length, multiply the thickness of material in hundredths of an inch by the constant whole number 44, and divide the product by the radius of the diameter of the flue in inches; the quotient will be the pressure allowable.

44 = C, constant.
T = thickness.
R = Radius of diameter in inches.
P = pressure allowable.

$$\text{Formula}: \frac{C \times T}{R} = P.$$

EXAMPLE.

Length of flue, 18 feet; diameter of flue, 14 inches; thickness of material, $\frac{31}{100}$:

Constant, 44.

$$\text{Then, } \frac{44 \times 31}{7} = 195 \text{ lbs., pressure allowable.}$$

For every foot or fraction thereof over 18 feet, deduct three pounds per square inch from the pressure allowable on a flue 18 feet in length or add $\frac{1}{100}$ of an inch to the thickness of material

required for a flue 18 feet in length for every 3 feet or fraction thereof over 18 feet.

To determine the thickness of material for any required pressure, multiply the pressure in pounds by the radius of the diameter of the flues in inches, and divide the product by the constant 44; the quotient will be the thickness required.

$$\text{Formula: } \frac{P \times R}{C} = T.$$

EXAMPLE.

Required the thickness of material for a flue 18 feet in length and 14 inches in diameter, requiring a working pressure of 195 pounds to the square inch:

$$\frac{195 \times 7}{44} = 31 +, \text{ thickness required.}$$

The thickness of lap-welded flues shall not be less than the product of the constant 2.20 multiplied by the diameter of the flues in inches, which will express the thickness in hundreths of an inch.

$$\text{Formula; } C \times D = T, \text{ minimum thickness.}$$

EXAMPLE.

Required the least thickness allowable for a lap-welded flue 14 inches in diameter:

$$2.20 \times 14 = \frac{30.80}{100}, \text{ least thickness allowable.}$$

Lap-welded flues 7 inches and not over 16 inches in diameter, over 5 feet and not over 10

feet in length, shall be re-enforced by one wrought-iron ring attached externally at the centre of the flue.

Lap-welded flues over 10 feet and not over 15 feet in length shall have two wrought-iron rings attached to the flue externally, equidistant between the ends of the flue, and there shall be attached one additional ring for every 5 feet or fraction thereof over 15 feet in length.

All such rings shall be good and substantially made, and properly and securely attached to the flues, and shall have a thickness of material of not less than the thickness of the material of the flues, and a width of not less than 2½ inches: *Provided, however*, Where such flues are made in lengths of not over 5 feet, and fitted one into the other, and substantially riveted, the wrought-iron rings may be dispensed with.

Lap-welded flues seven inches in diameter and less, shall be in accordance with the following table of thicknesses:

Diameter.	Thickness.	Diameter.	Thickness.	Diameter.	Thickness.
Inches.	*Inches.*	*Inches.*	*Inches.*	*Inches.*	*Inches.*
7 . . .	.165	6 . . .	.165	5	.148
4½ . .	.134	4 . . .	.134	3¾ . . .	.120
3½ . .	.120	3¼ . .	.120	3	.109
2¾ . .	.109	2½ . .	.109	2¼ . . .	.095
2 . . .	.095	1¾ . .	.095	1½ . . .	.083
1¼ . .	.072	1 . . .	.072		

10. For cylindrical boiler flues over 16 and less than 40 inches in diameter, the following formulas shall be used in determining the pressure allowable:

Let D = diameter of flue, in inches.
1760 = A, constant.
T = thickness of flue, in decimals of an inch.
P = pressure of steam allowable, in pounds.
$\frac{1760}{D} = F$, a factor.
$.31 = C$, a constant.

$$\text{Formula}: \frac{F \times T}{C} = P.$$

EXAMPLE.

Given a flue twenty (20) inches in diameter and thirty-seven one hundredths (.37) of an inch in thickness; required the pressure allowable by the inspectors:

$\frac{1760}{D} = F$, a factor; $\frac{1760}{20} = 88 = F$; then supplying values in the formula, $\frac{F \times T}{C} = P$, we have $\frac{88 \times .37}{.31} = P$. Performing the operation indicated, we have $\frac{88 \times .37}{.31} = 105 +$ lbs., pressure allowable.

CORRUGATED FURNACE-FLUES.

SEC. 10. The strength of corrugated flues, when used for furnaces or steam-chimneys (corrugation not less than 1½ inches deep), and provided that the plain parts at the ends do not exceed 6 inches in length, and the plates are not

less than $\frac{5}{16}$ thick, when new corrugated and practicably true circles, to be calculated from the following formula:

$$\frac{12500}{D} \times T = \text{pressure.}$$

T = thickness in inches.
D = mean diameter in inches.

EXAMPLE.

Given a corrugated flue 40 inches mean diameter, ½ inch thick; required the pressure allowed by inspectors:

$$P = \frac{12500}{D} \times T = \frac{12500 \times .5}{40} = \frac{6250.0}{40} = 156 \text{ pressure.}$$

11. The formulas for cylindrical lap-welded and riveted flues in boilers to be used as furnaces, which shall be used by inspectors in determining the pressure to be allowed, shall be as follows, viz:

Let D = diameter of flue in inches.
89600 = A, constant.
T = thickness of flue in decimals of an inch.
L = length of flue in feet, not to exceed 8 feet.
P = pressure of steam allowable, in pounds.

$$\text{Formula: } \frac{89600 \times T^2}{L \times D} = P.$$

EXAMPLE.

Given, a flue forty (40) inches in diameter, seven (7) feet in length, and five (5) tenths of an inch in thickness; required working pressure to be allowed:

Substituting values in the formula, and performing the operation indicated, we have—

$$P = \frac{89600 \times T^2}{L \times D} = \frac{89600 \times .25}{7 \times 40} = \frac{22400}{280} = 80 \text{ lbs. pressure}$$

Provided, That if rings of wrought-iron are fitted and riveted properly on, around, and to the flues, in such manner that the tensile strain on the rivets shall not exeeed 6,000 pounds per square inch of section, the distance between the rings shall be taken as the length of the flue in the formula.

EXAMPLE.

Given, a flue forty (40) inches in diameter, eight (8) feet long, and five (5) tenths of an inch in thickness, having one ring at the middle of its length; required the pressure allowable by the inspectors.

Substituting values in the formula, and performing the operation, we have—

$$P = \frac{89600 \times T^2}{L \times D} = \frac{89600 \times .25}{4 \times 40} = \frac{22400}{160} = 140 \text{ lbs. pressure.}$$

12. The feed-water shall not be admitted into any boiler at a temperature less than one hundred degrees Fahrenheit for low-pressure boilers, and one hundred and eighty for high-pressure boilers.

13. Whenever steamers use a pressure upon their boilers exceeding sixty pounds to the

square inch, they shall be inspected as high-pressure steamers and designated as such.

14. Vertical tubular boilers shall not be used on steamers navigating the Red River of the North and rivers whose waters flow into the Gulf of Mexico, unless the water-line is 2 inches above the upper end of the tubes and fire-line.

15. All steamers navigating rivers, having boilers externally heated, shall have a clear space of not less than six inches between the boilers and wood-work on either side, and four inches on the top of said boilers.

16. All steamers navigating the ocean, sounds, lakes, bays and rivers, the boilers of which shall be internally heated, shall have a clear space of at least four inches on either side, and at the top not less than two inches clear space above the covering of the boilers.

17. All boilers hereafter placed in steamers shall have a clear space of at least eight inches between the under side of the cylindrical shell and the floor or keelson.

All man-holes for the shell of boilers shall have an opening not less in diameter than 11 by 15 inches in the clear, except that boilers less than 34 inches diameter of shell have an opening in the clear, in man-holes, of not less than 9 by 14⅞ inches; all boiler shells between 34 and 38 inches diameter, an opening not less than 9 by 16 inches, and all boiler shells between 38

and 48 inches in diameter, an opening not less than 11 by 15⅞ inches.

18. All wood-work or other ignitible substance, approaching within two inches of the boiler, shall be suitably sheathed with metal, so adjusted as to permit a free circulation of air between the sheathing and the ignitible surface.

19. All boilers shall have a clear space at the back and ends thereof of two feet opposite the back-connection door. Slip-joints in steam-pipes shall, in their working parts, when the steamer is to be employed in navigating salt water, be made of copper or composition.

20. There shall be fastened to each boiler a plate containing the name of the manufacturer of the material, the place where manufactured, the tensile strength, the name of the builder of the boiler, when and where built.

21. Every sea-going steamer carrying passengers shall be supplied with an auxiliary or donkey boiler of sufficient capacity to work the fire-pumps.

22. All steamers shall have inserted in their boilers plugs of Banca tin, at least one-half inch in diameter at the smallest end of the internal opening, in the following manner, to wit: Cylinder boilers with flues shall have one plug inserted in one flue of each boiler; and also one plug inserted in the shell of each boiler from the inside, immediately before the fire line, and not

less than four feet from the forward end of the boilers. All fire-box boilers shall have one plug inserted in the crown of the back connection, or in the highest fire service of the boiler. All upright tubular boilers used for marine purposes shall have a fusible plug inserted in one of the tubes at a point at least two inches below the lower gauge-cock, and said plug may be placed in the upper head sheet when deemed advisable by the local inspectors. All fusible plugs, unless otherwise provided, shall have an external diameter not less than that of a one-inch gas or steam pipe screw-tap, except when such plugs shall be used in the tubes of upright boilers; plugs may be used with an external diameter of not less than that of a three-eighths of an inch gas or steam pipe screw-tap, said plugs to conform in construction with plugs now authorized to be used by this Board; and it shall be the duty of the inspectors to see that these plugs are filled with Banca tin at each annual inspection.

23. All steamers having one or two boilers shall have three suitable gauge-cocks in each boiler. Those having three or more boilers in battery shall have three in each outside boiler and two in each remaining boiler in the battery; and the middle gauge-cocks in all boilers shall not be less than four inches above the top of the flues, tubes, or crown of the fire-box.

24. Lever safety-valves to be attached to

marine boilers shall have an area of not less than one square inch to two square feet of the grate surface in the boiler, and the seats of all such safety-valves shall have an angle of inclination of 45 degrees to the center line of their axis.

The valves shall be so arranged that each boiler shall have one separate safety-valve, unless the arrangement is such as to preclude the possibility of shutting off the communication of any boiler with the safety-valve or valves employed. This arrangement shall also apply to lock-up safety-valves when they are employed.

Any spring-loaded safety-valves constructed so as to give an increased lift by the operation of steam, after being raised from their seats, or any spring-loaded safety-valve constructed in any other manner, or so as to give an effective area equal to that of the aforementioned spring-loaded safety-valve, may be used in lieu of the common lever-weighted valve on all boilers on steam vessels, and all such spring-loaded safety-valves shall be required to have an area of not less than one square inch to 3 square feet of grate surface of the boiler, and each spring-loaded valve shall be supplied with a lever that will raise the valve from its seat a distance of not less than that equal to one-eighth the diameter of the valve-opening, and the seats of all such safety-valves shall have an angle of inclination to the center-line of their axis of 45 degrees.

But in no case shall any spring-loaded safety-valve be used in lieu of the lever-weighted safety-valve, without first having been approved by the Board of Supervising Inspectors.

The first paragraph of this section applies to valves constructed in material, workmanship and principle according to the drawings for a safety-valve printed with these rules, and all common lever safety-valves to be hereafter applied to the boilers of steam vessels must be so constructed.

When this construction of a safety-valve is applied to the boilers of steamers navigating rough waters, the link may be connected direct with the spindle of the valve: *Provided always*, That the fulcrum or points upon which the lever rests are made of steel, knife or sharp edged, and hardened; in this case the short end of the lever should be attached directly to the valve-casing. In all cases the link requires but a slight movement, not exceeding one-eighth of an inch.

All the points of bearing on lever must be in the same plane.

The distance of the fulcrum must in no case be less than the diameter of the valve-opening.

The length of the lever should not exceed the distance of the fulcrum multiplied by ten.

The width of the bearings of the fulcrum must not be less than three-fourths (3/4) of one inch.

The length of the fulcrum link should not be less than four inches.

The lever and fulcrum link must be made of wrought-iron or steel, and the knife-edged fulcrum points and bearings for the points must be made of steel and hardened.

The valve, valve-seat, and bushings for the stem or spindle must be made of composition (gun-metal) when the valve is intended to be attached to a boiler using salt water; but when the valve is to be attached to a boiler using fresh water, and generating steam of a high pressure, the parts named, with the exception of the bushings for the spindle, may be made of cast-iron.

The valve must be guided by its spindle, both above and below the ground seat and above the lever, through supports either made of composition (gun-metal) or bushed with it.

The spindle should fit loosely in the bearings or supports.

When the valve is intended to be applied to the boilers of steamers navigating rough waters, the fulcrum-link may be connected directly with the spindle of the valve; providing always, that the knife-edged fulcrum points are made of steel and hardened, and that the vertical movement of the valve is unobstructed by any lateral movement.

In all cases the weight must be adjusted on the lever to the pressure of steam allowed in each case by a correct steam-gauge attached to the boiler. The weight must then be securely fastened in its position and the lever marked, for the

purpose of facilitating the replacing of the weight should it be necessary to remove the same; and in no case shall a line or any other device be attached to the lever or weight except in such a manner as will enable the engineer to raise the valve from its seat.

Donkey boilers used on all steam vessels for driving pumps, hoisting engines, electric lights, or other purposes, must be inspected the same as the main steam-boilers, and supplied with water and steam gauges, and the safety-valves must comply with the same regulations as the main boilers.

The area of all openings in boilers and connections leading from boilers to safety-valves, both thc lever and spring-loaded valves, used on marine boilers, shall not be less than the area of the valve used in said safety-valve.

25. All steam-gauges heretofore in use on steamers shall be admissible by the inspectors, and other steam-gauges hereafter made, of equal merit, shall be allowed.

26. All boilers or sets of boilers shall have attached to them at least one gauge that will correctly indicate a pressure of steam equal to 80 per cent. of the hydrostatic applied by the inspectors.

27. The appliances in use on steamers constructed prior to the 28th of February, 1872, for determining the height of water in the boilers, shall be considered reliable low-water gauges.

28. There must be means provided in all boilers using the "low-water gauges," which are operated by means of a float inside the same, to prevent the float from getting into the steam pipe.

29. In applying the hydrostatic test to boilers with a steam-chimney, the test-gauge should be applied to the "water-line" of such boilers.

30. All horizontal cylindrical boilers used on steamers navigating the waters flowing into the Gulf of Mexico shall be provided with a reliable low-water gauge.

31. The hydrostatic test applied to the boilers of towing freight-boats on the Mississippi River and its tributaries shall be in the proportion of one hundred and fifty (150) pounds to one hundred pounds working steam pressure allowed; and the inspectors shall test all such boilers on said steamers for the amount of steam allowed.

STEAM PRESSURE ALLOWED ON BOILERS.

[32.] Thickness of iron.	34 inches diameter.	36 inches diameter.	38 inches diameter.	40 inches diameter.	42 inches diameter.	44 inches diameter.	46 inches diameter.
Inches.	*Pounds*	*Pounds*	*Pounds*	*Pounds*	*Pounds*	*Pounds*	*Ponnds*
.19	140.82	133.	126.	119.70	114.	108.81	104.08
.20	148.23	140.	132.63	126.	120.	114.54	109.56
.21	155 64	147.	139.26	132.30	126.	120.27	115.04
.22	163.05	154.	149.50	138.60	132.	126.	120.52
.23	170.47	161.	152.52	144.90	138.	131.72	126.
.24	177.88	168.	159.15	151.20	144.	137.45	131.47
.25	185.29	175.	165.79	157.50	150.	143.18	136.95
.26	192.70	182.	172.42	163.80	156.	148.80	142.43
.27	200.11	189.	179.05	170.10	162.	154.63	147.91
.28	207.53	191.	185.68	176.40	168.	160.36	152.5[illegible]
.29	214.94	203.	192.31	182.70	174.	166.04	158.87
.30	222.35	210.	198.94	189.	180.	171.81	164.34
.31	229.76	217.	205.57	195.30	186.	177.54	169.82

LICENSED OFFICERS.

1. Before an original license is issued to any person to act as a master, mate, pilot, or engineer, he must personally appear before some local board or a supervising inspector for examination; but upon the renewal of such license, when the distance from any local board or supervising inspector is such as to put the person holding the same to great inconvenience and expense to appear in person, he may, upon taking the oath of office before any person authorized to administer oaths, and forwarding the same, together with the license to be renewed and Government fee, to the local board or supervising inspector of the district in which he resides or is employed, have the same renewed by the said inspectors, if no valid reason to the contrary be known to them; and they shall attach such oath to the stub end of the license, which is to be retained on file in their office. And inspectors are directed, when licenses are completed, to draw a broad pen and red ink mark through all unused spaces in the body thereof, so as to prevent, so far as possible, illegal interpolation after issue.

2. The classification of engineers on the lakes and seaboard shall be as follows:

Chief engineers of ocean steamers.

Chief engineers of lake, bay and sound steamers.

Chief engineers of river steamers.

First assistant engineers.

Second assistant engineers.

Third assistant engineers.

Special engineers.

All steamers of over one hundred tons burden shall carry at least one chief engineer.

First assistant engineers may act as first assistants on any steamer.

Second assistant engineers may act as first assistants on steamers of seven hundred and fifty tons and under, and second assistants on any steamer.

Third assistants may act as second assistants on steamers of 750 tons and under, and third assistants on any steamer.

Special engineers may be assigned to act in any capacity for which they are qualified on steamers of one hundred tons and under.

Inspectors may designate upon the certificate of any chief engineer the tonnage of the vessel on which he may act, and they may also designate any assistant engineer as special engineer on steamers of one hundred tons or under, and may restrict an engineer to a particular vessel.

3. Engineers on high pressure steamers navigating rivers shall be designated as chief engineers (H. P.), assistant engineers (H. P.), and special engineers (H. P.).

Assistant engineers may act as chief engineers

on high-pressure steamers of one hundred tons burden and under, of the class and tonnage, or particular steamer, for which the inspectors, after a thorough examination, may find them qualified. In all cases where an assistant engineer is permitted to act as first [chief] engineer, the inspector shall state on the face of his certificate of license the class and tonnage of steamers, or the particular steamer, on which he may so act.

4. It shall be the duty of an engineer, when he assumes charge of the boilers and machinery of a steamer, to forthwith thoroughly examine the same, and if he finds any part thereof in bad condition, caused by neglect or inattention on the part of his predecessor, he shall immediately report the facts to the local inspectors of the district, who shall thereupon investigate the matter, and if the former engineer has been culpably derelict of duty, they shall suspend or revoke his license.

5. No original license shall be issued to any person to act as engineer, except for special license on small pleasure steamers, who cannot read and write, or who has not served at least three years in the engineer's department of a steam-vessel, or as a regular machinist in a machine works; provided that any person who has served for a period of three years as a locomotive or a stationary engineer may be licensed to act as engineer on steam vessels after having

had not less than one year's experience in the engineer's department of a steam vessel.

CHANGES IN STEAMBOAT RULES.

Amendments Made by the Supervising Inspectors and Approved.

Section 5, rule 5, has been amended so that no person shall receive an original license as engineer who has not served at least three years in the engineer's department of a steam vessel, except locomotive and stationary engineers, and a regular steam engine machinist; and no person shall receive a license who is not able to determine the weight necessary to be placed on the lever of a safety-valve to withstand any given pressure of steam in a boiler. By section 18 the grade of an engineer or pilot shall not be raised during the term for which the license was granted, except by consent of the board that granted the license.

CHAPTER XVI.

BURNING OF FUEL.*

It being agreed that heat is the agent which does work in an engine, and that steam, air, and vapor are but the instruments for transmitting the motion of heat to the machinery, our object will be to store up in an elastic working substance the heat derived from fuel, and to guard against loss as far as possible.

As a general rule *chemical ccmbination* is accompanied by the evolution or production of heat, and chemical decomposition by the disappearance of heat equal in amount to that produced during the previous combination of the elements which are undergoing separation.

Combustion, or burning, is the name given to rapid chemical combination attended with the evolution of intense heat.

It is necessary to bear these facts in mind in estimating the heating effect of fuel. Thus, where hydrogen and oxygen exist in coal in the proportion necessary for forming water (viz: one of hydrogen to eight of oxygen by weight,) it is usual to assume that they do not influence the heat of combustion.

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* Goodeve.

The hydrogen is taken to have been already burnt in oxygen. In coal there may be 5 per cent. of hydrogen, and 4 per cent. of oxygen: this would leave 4½ per cent. of hydrogen available for heating purposes. There appear to be exceptions to the above rule, and Dr. Percy gives the result of an experiment where two coals closely agreeing in ultimate composition have been found to differ by 5 per cent. in heating power.

The composition of various kinds of coal is given by Dr. Percy, in his work on fuel, and it is well known that the differences in the constituent parts of coal are very great, and give rise to qualities of various kinds which influence the selection to be made for heating purposes.

The heat given out in the burning of hydrogen and carbon is estimated as follows:

	Units of Heat.
1 pound of hydrogen consumes about 36 pounds of air and gives out.	62,032.
1 pound of carbon burnt to carbonic oxide, about 6 pounds of air and gives out. . .	4,452.
1 pound of carbon burnt to carbonic acid, about 12 pounds of air and gives out . .	14,545.

According to Dr. Percy the *heating power* of a substance is the number of units of heat produced by the combustion of a unit of weight of the substance; and if the unit of heat be defined according to the Centigrade scale

Heating power of hydrogen is 34,462.

Heating power of carbon is 2,473, when burnt to carbonic oxide.

Heating power of carbon is 8,080, when burnt to carbonic acid.

Also 1 pound of hydrogen evaporates 64.2 pounds of water at 212° Fah.

Also 1 pound of carbon (burnt to carbonic oxide) evaporates 4.6 pounds of water at 212° Fah.

Also 1 pound of carbon (burnt to carbonic acid) evaporates 15 pounds of water at 212° Fah.

It does not appear that the absolute heat of combustion can be increased, but it is easy to pile up the waves of heat in an enclosed space, and thereby to increase wonderfully the apparent power of the combustion.

The furnace may be looked upon as a large chemical apparatus in which coal and air are to be mixed together in the proportion best adapted for burning the fuel without waste. In performing this operation, an engineer falls very far behind a scientific chemist when operating on a small scale in his laboratory.

Thus a chemist in burning one pound of ordinary coal in a carefully protected chamber, would cause the heat from the fuel to evaporate (say) 14 pounds of water, whereas the evaporation per pound of coal in a steam boiler seldom exceeds 10 pounds, or 10½ pounds, of water, a common performance being the evaporation of from 6 pounds to 8 pounds of water.

Looking at the question as one of admixture of fuel and air, the rough numbers usually

quoted on the authority of Rankine are the following: For the actual burning of ordinary coal in a furnace, 12 pounds of air are required in order to combine with the constituents of each 1 pound of coal.

But the gaseous products of combustion must be largely diluted, otherwise the air would not get at the fuel, and for this dilution as much air again is required, making a supply of 24 pounds of air to each, pound of fuel.

Thirteen cubic feet of air, at 60° Fah. and under a pressure of 30 inches of mercury, weigh abont 1 pound. Therefore 312 cubic feet of air are required for each 1 pound of fuel, which comes to nearly 700,000 cubic feet of air for the effective burning of one ton of coal.

That gas and hydrocarbon vapor proceeding from coal require a good supply of air for burning was frequently shown by Faraday in a lecture experiment, and his illustration goes to the substance of the whole matter.

The device was to soak a little cotton-wool in any hydrocarbon liquid, and set it on fire in a jar of oxygen gas. In such a case the hydrogen devours the oxygen and the flames light up with dazzling brilliancy, but very soon the supply of oxygen fails, the light becomes less, when all at once, for no apparent reason, the burning wool throws out a dense mass of black flakes, which fill the jar in a thick cloud.

The quantity of soot ejected would surprise any one but a chemist, as few would be aware that the unburnt liquid was capable of throwing out such a supply of carbon.

It is needless to say that the effect here produced in the jar of oxygen is the same as that occurring in the chimney of a steam boiler when the supply of air is defective, the result being that so frequently seen, viz.: the pouring out of dense black smoke into the atmosphere. The loss of heat from unburnt gases may also take place without being made evident by the issuing of smoke. Thus carbouic oxide may pass away instead of carbonic aid.

There have been a great number of inventions relating to the prevention of smoke in steam boilers, which cannot be discussed in the space here available.

Smoke once formed cannot be burned. The proper thing to do is to devise means to prevent the formation of smoke.

The various modes in which fuel is wasted have been classified by Rankine somewhat as follows:

1. Fuel is lost by the escape of gases in an unburnt state, or by permitting black smoke to be thrown off.

Here the supply of air is defective, and the physical action is traced to the remarkable affinity of hydrogen for oxygen gas; whereby the

oxygen is absorbed to the exclusion of carbon in the first instance.

2. There is waste from external radiation and conduction. M. Peclet states that the quantity of heat radiated from incandescent charcoal is 5 per cent. of the total heat of combustion, and that the heat radiated from coal somewhat exceeds that radiated from charcoal. The practical conclusion to be drawn from this statement is, that the heat radiated from the burning fuel should be carefully intercepted in every direction.

Hence the economy resulting from the use of an internally fired boiler with internal furnace tubes.

As to the heat radiated into the ash-pit, that is carried back again to the fire by the current of entering air. In respect of the loss of heat by conduction, that is obviated as much as possible by the use of fire-brick; and where the furnace is outside the boiler, the resistance to conduction is increased by double layers of brickwork with enclosed air spaces between the layers.

3. There is loss of heat by the escape of gases up the chimney at a temperature above that which is necessary for maintaining the draught.

A general idea of the value of a chimney in promoting the draught of a fire may be gathered from a statement of a law which appears to be approximately true, viz: That the velocity of air,

as due to increased pressure, is that acquired in falling down a height equal to the uniform column which gives the increased pressure.

In making any calculation on this subject it is usual to adopt the hypothesis that air is incompressible and behaves as a liquid.

Let the increase of pressure support 5 inches of water. We know that 29.922×13.596 inches of water balance the pressure of the atmosphere which would be produced by a stratum of incompressible air 26,214 feet high.

Therefore, 1 inch of water will balance 64.4 feet of air.

Hence 5 inches of water balances 322 feet of air: therefore velocity due to increase of pressure

$$= \sqrt{64.4 \times 322}$$
$$= 64.4\sqrt{5}$$
$$= 144 \text{ feet per second, very nearly.}$$

According to the old rule the area of the chimney should be $\frac{1}{10}$ that of the fire-grate, and there should be 1 square foot of fire-grate for each horse-power.

Rankine gives formula for computing the height of a chimney in order to produce a given draught, and states that the best chimney draught takes place when the absolute temperature of the gas in the chimney is to that of the external air as 25 to 12, or when the density of the hot gas is one-half that of the external air.

For example, if the temperature of the external air be 50° Fah., the best temperature of the hot gas in the chimney will, according to this rule, be 602° Fah., which is less than that of melting lead, viz., 620° Fah.

Hence the rule that to insure the best possible draught in a chimney the temperature of the hot gas should be nearly sufficient to melt lead.

If the temperature of the furnace itself be estimated at 2,400° Fah. and that of the issuing gases at from 600° Fah. to 700° Fah. or even higher, as is often the case, we see that 25 per cent. of the heat of combustion passes up the chimney and is consumed in producing a draught of air through the furnace grate.

The loss of heat from the waste gases may be lessened by the use of an economizer for heating the feed-water.

Sir W. Fairbairn, in his treatise on "Mills and Millwork," describes an economizer introduced by Mr. Green, of Wakefield, as consisting of a series of upright tubes forming a supplementary boiler placed in the main flues, and states that the formation of soot on the pipes was the source of the ill success of previous attempts in this direction.

This difficulty has been overcome by an apparatus of scrapers or cleaners, and it is found that when the waste gases escape at a temperature of 400° Fah. or 500° Fah. the feed-water

can be heated to an average of 225° Fah., the temperature of the gases after leaving the pipes being reduced to 250° Fah.

To produce this effect 10 square feet of heating surface should be provided for each horse-power.

4. Fuel is wasted by brittleness, dust and small pieces dropping unburnt through the bars into the ash-pit.

5. The fuel is rendered less effective by the presence of earthy compounds, which form clinkers, abstract heat uselessly, and choke up the grate.

CHAPTER XVII.

FORMS OF STEAM BOILERS.

THE construction of a boiler should be regarded from two points of view: (1) There is the general form and structure adapted to support the bursting pressure of the steam. (2) There are considerations arising from the unequal action of the heat of the burning gases, and there are precautions to be taken for diminishing the waste of heat.

Considerations which influence the forms of boilers.—The early boilers were designed in simple defiance of all mechanical principles. Without doubt the safety of a boiler depends on the strength of the metal, but it is quite wrong to say that it is independent of the form of the shell, and any one who thinks for a moment on the subject will comprehend that a cylindrical tube of some kind is the proper vessel wherein to retain a supply of steam under pressure. The strongest form of a vessel for holding a gas under pressure is a sphere: that is the natural form for the purpose, as we learn in blowing a soap-bubble.

But no one would recommend a spherical

boiler, there are so many practical reasons against it.

The next best theoretical form is a cylinder, which may be of any size, and only suffers from weakness at two ends.

With cylindrical boilers there is a difficulty about strengthening the ends.

Theory would tell us to make the ends hemispherical, when their strength would be as much as twice that of the cylindrical sides. There is, however, a practical objection to forming a hemispherical or egg-end, as it is called, from the boiler plate; and the result is that the ends are commonly made of flat plate, which is strengthened artificially in such a manner as to give security against danger. As to the strength of boiler-plate, the general rule is that a rolled plate is less strong per square inch of section than a thick bar of the same iron.

Also the reduction of strength is more marked in the transverse than in the longitudinal direction. There is a further subtraction from the strength by riveting; and, according to Sir W. Fairbairn, the breaking strains of riveted joints of boiler plate are estimated somewhat as follows:

	Pounds.
Iron	50,000
Double riveted joints	35,000
Single riveted joints	28,000

Per square inch of section.

Since the plates after rolling are stronger in the direction in which they are rolled than transversely, it is apparent that they should be so disposed that the direction of greatest strength should encounter the greatest strain.

But the strain on the longitudinal section is greater than that on the transverse, and hence the plates are wrapped round the circumference.

As a consequence of the theoretical disproportion between the two strains, *it is further recommended that the longitudinal seams should be double-riveted with ¾-inch rivets, pitched about 2½ inches longitudinally and 2 inches diagonally.*

The transverse seams are single-riveted, the pitch being 2 inches. To double-rivet them would appear to add but little to the strength of the boiler, though it would increase its weight and cost.

The tubes diminish the area of the flat ends and relieve the pressure tending to rupture the material on a transverse section. They further act as stays for holding the ends together.

On both these accounts a boiler with internal tubes becomes stronger than it would be without the tubes, so far as transverse rupture is concerned. The flat ends remain, however, a source of anxiety, and in order to support them gusset stays and tie rods are employed, as to

which something will be said after the disturbing action of heat has been noticed.

EFFECT OF HEAT.

The wear and ultimate strength of a boiler are greatly influenced by adopting a construction which shall provide for the inevitable changes of form caused by unequal expansion and contraction due to changes in temperature.

Heat is motion, and as soon as the fire inside a furnace flue is lighted, the metal on the top becomes more heated than under the surface, and the tube arches itself in consequence of the greater expansion of the hotter portion.

And not only so, but the flue lengthens as a whole, and the flat ends bulge outwards. Finally the water becomes heated and the whole structure elongates, and unless sufficient allowance be made for the pulsating movements, straining will occur, which may possibly end in rupture.

The linear expansion of wrought iron (soft forged) under the action of heat is stated to be .0012204 for a rise in temperature from 0° C. to 100° C.

Thus a bar of iron 30 feet long expands about $\frac{7}{10}$ inch for a rise in temperature of 132.2° C.

The expansion of the parts of a boiler as caused by heat is of course capable of accurate measurement; and in particular the so-called

hogging of a boiler flue has been observed by applying three gauge rods attached at equal distances along the crown of the tube; each rod is carried vertically upwards, and passes through a stuffing box in the shell of the boiler, whereby it has been possible to observe very accurately the distortion of the flue.

One boiler experimented on was 28 feet long, and it was found that the flue rose 3/8 inch when the flame passed around the boiler in the ordinary way along the side flues, and that it rose 1/2 inch when the flame was carried directly into the chimney without heating the outer shell.

The gauge-rod at one-fourth the length of the boiler from the front end rose as much as that, and in one case 1/16 in. more.

Also the colder the water at starting the greater was the distortion, and generally the action was more severe just after the lighting of the fires. As soon as the whole of the water became permanently heated the gauge-rods retired to their primary position, the distortion of the flues seldom lasting for more than an hour. Mr. Fletcher recommends that the end plates of boilers, to be used at a pressure of 75 pounds per square inch, should be 1/2 inch in thickness, increasing to 9/16 inch for increased pressure within moderate limits, excessive thickness being undesirable, as confining or restraining the necessary movement of the furnace tubes. The object

is to strengthen the end plate, and yet to preserve its elasticity; and in carrying out this intention it is a rule to attach the plate at the front of the boiler to the shell by external angle iron. This mode of construction is not, however, adopted at the opposite end.

The furnace flues are a vulnerable part of the boiler, inasmuch as they are liable to yield by collapsing unless sufficiently strengthened.

The subject of strengthening the internal tubes of the internal fire-flue boiler was investigated by Sir W. Fairbairn, whose experiments led to the following conclusions:

(1) The strength of a tube to resist collapse by external pressure is inversely as its diameter.

(2) The strength varies inversely as the length.

(3) The collapsing pressure in pounds per square inch

$$=806300 \times \frac{(\text{Thickness of plate in inches.})^{2.19}}{\text{Length in feet} \times \text{diameter in inches.}}$$

In these experiments the ends of the tubes were firmly attached to rigid plates, and the vessel in which the compressing force was applied was a cast-iron cylinder 8 feet long, 28 inches in diameter, and 2 inches thick, which could be safely strained as far as 500 pounds per square inch.

Into this cylinder air was forced by a pump,

and produced any required pressure on the surface of a quantity of water which nearly filled the cylinder, and in which the tube under trial was completely immersed. There were two gauges for reading the pressure, and a safety valve in addition, which was loaded by a weight.

The experiments made by Sir W. Fairbairn were valuable as calling attention to a material subject connected with the construction of boilers; but it appears that a special contrivance

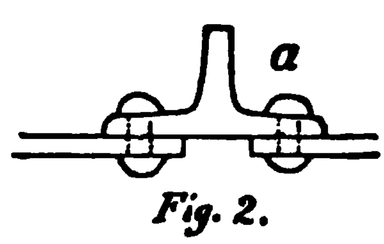

FIG. 41.

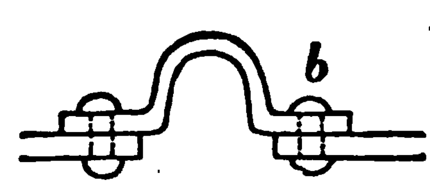

FIG. 42.

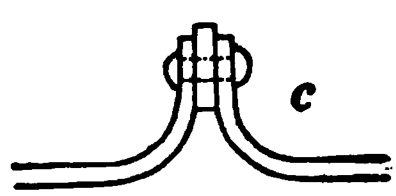

FIG. 43.

for strengthening furnace flues, while allowing at the same time for their expansion and contraction, had been originated some years prior to the researches to which reference has been made. At a meeting of the Institution of Mechanical Engineers held in 1876, Mr. Adamson stated that he had first employed a "flanged seam," as it is termed, for the strengthening of furnace flues, as early as the year 1851.

The Figs. 41, 42 and 43, exhibit three forms of joints as applicable to tubes subjected to external pressure.

The first, marked *a*, consists of a ring of T-iron, riveted as shown. It is abundantly strong, and is in a form which has been adopted for centuries past in strengthening guns. The weakness of a tube to resist a bursting pressure on a longitudinal section has been already demonstrated, and a common method of strengthening it has been to apply parallel rings at intervals along its length.

In this way steam cylinders of great length, which are subjected alternately to a bursting and compressing pressure, have been strengthened. Every one is aware of the accession of strength caused by the flange of a pipe. Since action and reaction are equal and opposite, we might have anticipated that a form of tube constructed so as to withstand a bursting pressure from within, would also be the form best adapted for resisting a collapsing pressure from without.

The difficulty of calculation in the latter case arises from the liability to deformation, which is soon set up, after which theory is of little use in enabling us to predict a result.

But although the joint *a* has ample strength, it is deficient in another quality which is of importance, viz., it does not permit any alteration of length. The whole furnace tube is rigid, and expands or contracts as one piece.

Whereas, in the joint marked *b*, the expansion or contraction of each length of the tube is provided for by the arched or the corrugated piece, and here there is increased strength combined with power of expanding or contracting freely. In the joint *c*, which is known as "Adamson's flanged joint," there is the strength of the T-iron directly combiued with the curved end, which allows of unimpeded expansion or contraction.

The arrangement is most convenient and effective, and is particularly valuable as giving a seam where the rivets are protected from the furnace gases, and are, in fact, immersed in water, one consequence of the construction being that the joint will bear intense heat much better than any other where the rivets are exposed.

STRENGTH OF BOILERS.

While the pressure acts from the *centre* radially out to the circumference on an indefinite number of radial lines, the mathematics of the strength of the shell supposes that the pressure acts as the arrows *a*, *a*, *a*, *a*, in Fig. 45, that is, tending to lift the top from the bottom. In Fig. 44 the pressure represented by arrow *a* is resisted by arrow *a'*, and similarly arrow *b* and *b'* resist each other, and as each is equal, we may assume any one direction as that in which the pressure acts, as *a*, *a*, *a*, *a*, in Fig. 45, tending to

part the boiler through the line *b*, *b*. We might, as stated above, assume any other direction of the pressure, but as the shell is as strong through *b*, *b* as it is through any other similar section, the strength as ascertained at *b*, *b* will be the strength of the boiler. The standard of comparison between different brands of boiler iron is based on the strength of a *square* bar of that iron, the sides of which are one inch in length, and a section through the bar representing one *square* inch. The tensile strength of such a bar will average 50,000 pounds. Assuming, then, that the pressure tends to tear the top from the bottom of the shell, it is apparent that the force acting to do this is represented by the length of the shell in inches, multiplied by the diameter in inches and by the pressure per square inch. The iron being the same thickness throughout the shell, we will also assume that a section one inch in length of the boiler is a *fac-simile* as to strength of any other section of similar length. The iron being $\frac{5}{16}$ of an inch thick and one inch (assumed) in length, it is plain that its strength is but $\frac{5}{16}$ of our standard of comparison, 50,000 (or 15,625 pounds). Now, as we have *two* sides of the boiler of $\frac{5}{16}$ thick, we must double the above amount, which = 31,250 pounds, as the strength of our section if it were without a joint. The strength of a single-riveted seam (Fig. 46) has been ascertained to be

but 56 per cent. of the solid iron, and hence we must take that percentage of 31,250 to arrive at the strength of the weakest portion, which, of course, comes under the principle that the strength of any structure is equal only to its weakest part. 56 per cent. of 31,250 = 17,500, which is the product of the length of our section (1 inch) × by the diameter, say 48 inches, multiplied by the pressure per square inch necessary to burst the boiler. We have, therefore, in order to find the pressure per square inch necessary to burst the boiler, only to divide 17,500 by the diameter (the length being 1 inch or unity). This gives us nearly 365, which, divided again by our factor of safety, we will take as 6 (or the working pressure = to $\frac{1}{6}$ the bursting pressure), and we obtain 60 nearly, as the proper working pressure.

The strength of a joint, like Fig. 46, through the rivet holes, may be found by taking the length *a*, less the diameter of one hole multiplied by the number of holes. This leaves the length of solid iron between the holes, which, multiplied by the thickness and by the strength of a sectional inch, and from 15 to 20 per cent. deducted from the result, for injury done the iron by the punch, leaves the probable strength of metal between the holes. A staggered or double-riveted joint, shown in Fig. 53, has 70 per cent. of the strength of the solid sheet, as as-

21

certained by Fairbairn's experiments. From inspection, it will be seen that there is as much metal between the holes *h*, *c* and *h*, *f* as there is between *c*, *f*. The metal between the holes is much greater than in Fig. 46, while the number of rivets is in proportion to the metal left between the holes, and hence the strength of the joint may be deducted from the metal left between the holes on a line, *a*, *b*. The strength of a rivet, to resist shearing, is about 50,000 lbs. per square inch, hence the sectional area of one rivet, multiplied by the number of rivets, and again multiplied by 50,000, will closely approximate their shearing or detensive strength. The rivet area strength should be kept as nearly equal as possible to the strength of sheet between the holes. The strength of the sheet to resist crushing is represented by Fig 47. It is plain that if the rivet does not shear, nor the sheet rupture between the holes, the part *a* must be pushed or sheared out of the way, if rupture takes place. The resistance which the part of piece *a* offers is represented by the area *b*, *b*, or the length *d*, by the thickness of the plate plus the equal area *c*, *c*, and the whole multiplied by the detensive strength of the sheet, which is about 50,000 lbs. per square inch. In designing a joint, the piece *a*, Fig. 47, the shearing strength of the rivets and the strength between the holes should be as nearly equal as possible, to make

as strong a joint as is possible. The joint shown in Fig. 51 has a welt piece *a*, added, and is much used in boiler work. It does not add as

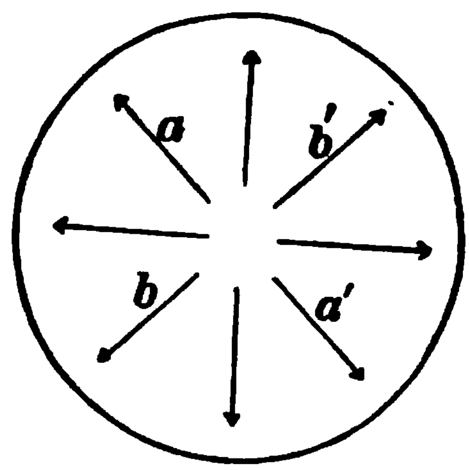

FIG. 44.

much strength as is commonly accredited to it. The line of rivets *d* and *f*, Fig. 48, is commonly spaced twice as far as the middle one *e*, and therefore presents only half the shearing

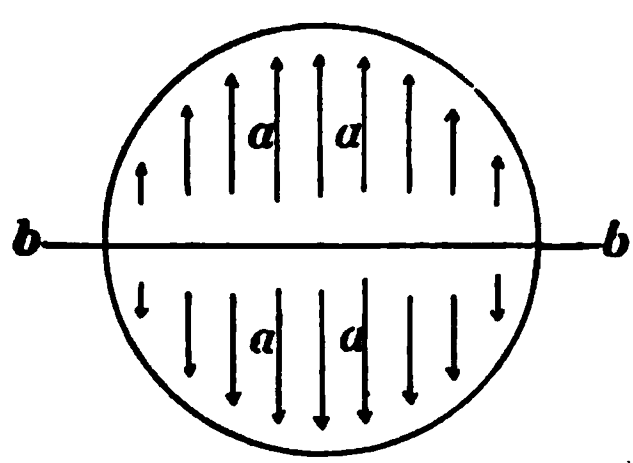

FIG. 45.

strength of rivets that line *e* does, and hence if rupture were to take place it would occur first through either *d* or *f*, if it were not for the stronger joint *e*. Therefore, after the internal

pressure exceeeds the strength of joint *d* or *f*, they are no longer of use, as rupture is alone prevented by the superior strength of the joint *e*. The welt strengthens the joint simply because it puts the rivet *g*, Fig. 51, into double shear, that is, the rivet must be sheared through the line *h* and *i*, and therefore presents double resistance. The part of the plate in front of the rivet must be proportionally increased to meet the increased resistance, and the pitch of the holes increased as much as possible and still retain a tight joint. A combination of this joint and the staggered

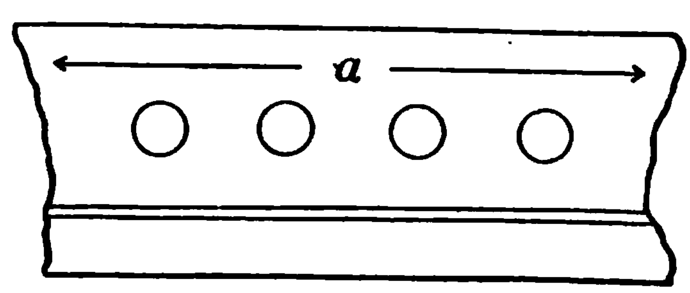

FIG. 46.

riveted joint would make the strongest possible longitudinal joint, as it would give the greatest amount of metal between the holes, and also present the rivet in double sheet. Fig. 49 shows a mode of making a joint by which the strength might be made equal to that of the solid sheet. The distance between the holes, by making the joint long enough, might be made equal to the width of the sheet, or in excess of it, for injury done in punching. The joint shown in Fig. 52 is not strengthened by the second row of rivets,

barring the grip due to contraction of the extra rivets, as the strength of metal between either row is the same, as is also the number of rivets,

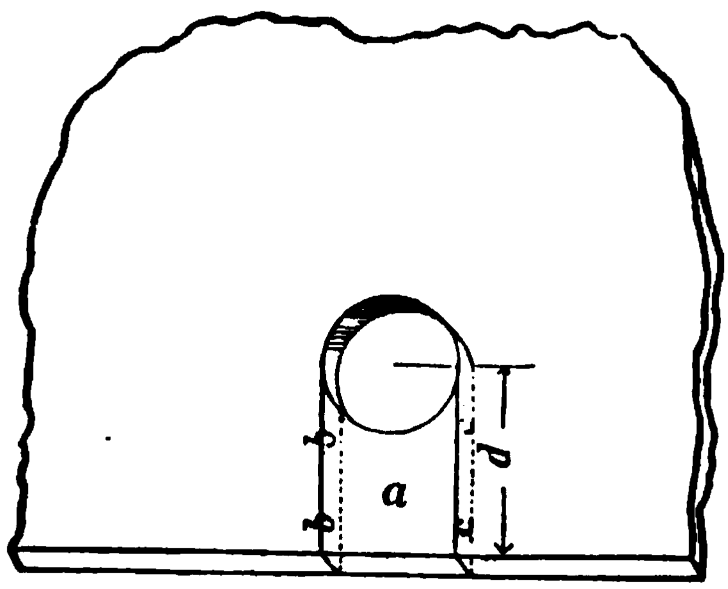

FIG. 47.

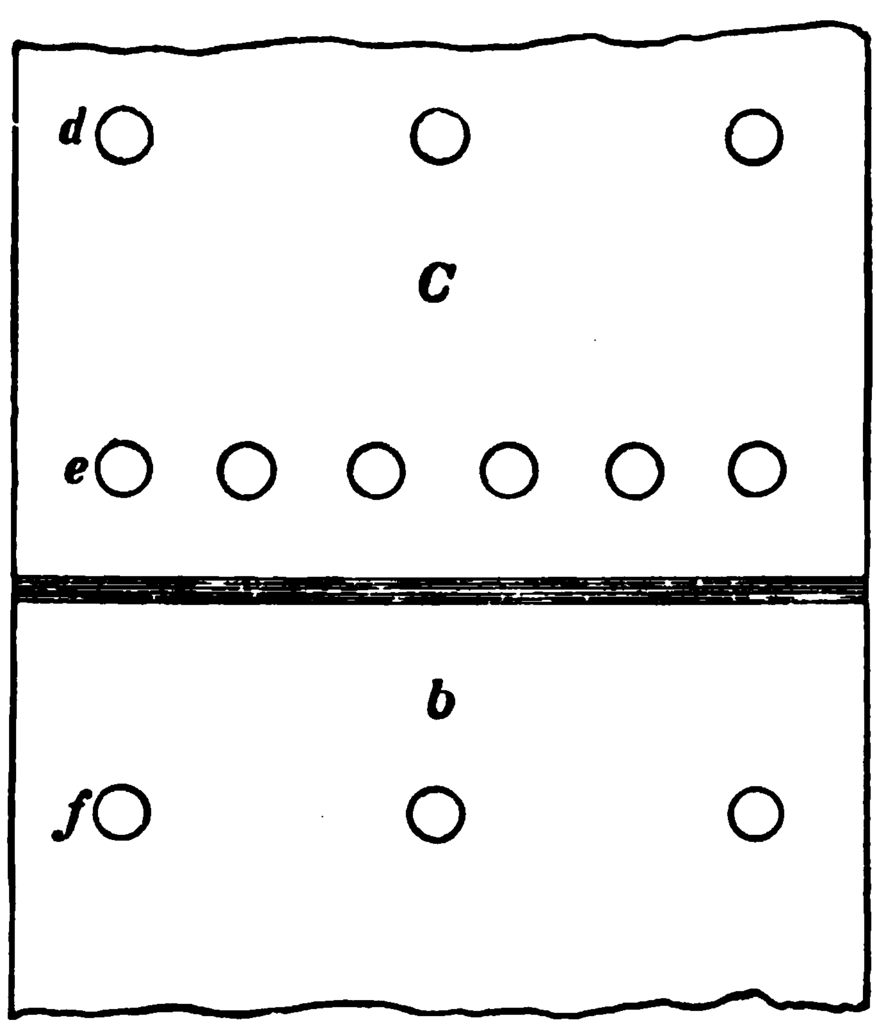

FIG. 48.

etc.; and the joint is as weak through one row of rivets as the other. We have been considering the longitudinal strength only. The transverse strength of a boiler is that which prevents one end from being torn from the other. This

FIG. 49.

FIG. 50.

strength is represented by the thickness of the iron × by the circumference × by the strength of a sectional inch, and the whole less the percentage of loss due to the kind of joint. However, a single rivet joint will suffice, generally,

as the boiler is about twice as strong transversely as longitudinally.

Persons buying boilers would do well to have samples of the iron, from which the boiler is to be built, tested for tensile strength, and take no interested parties' word, or bond either, on that point. An actual test settles the matter beyond doubt. All holes should be *drilled* and rounded in the inner edge, as shown at *a*, *a*, *a*, *a*, Fig. 50, as a drilled hole, if not rounded, will shear the rivet sooner than a punched hole. Flat surfaces should be stayed to from six to ten times the pressure they are to resist. The *area* of rivets holding crow-feet to boiler heads, etc., should be equal to the stay coupled to said crow-foot. It is a good plan to make stays coupled to crow feet slightly shorter than the required length. Expand them by heat, and, on cooling, they will all be found tight and necessarily bearing their proportional strain. It is a *bad* plan to collect one end of stays (such as crown sheet stays when stayed to shell) into a smaller area than the other, as it is plain to be seen that the smaller area must resist more than its share in proportion as its area is smaller.

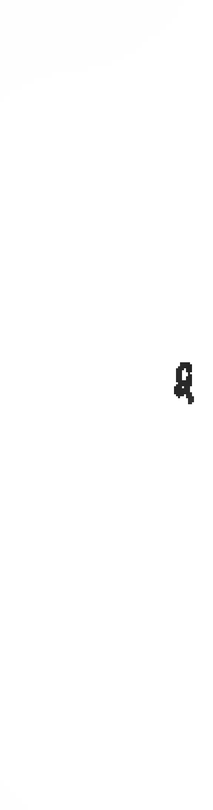

FIG. 51.

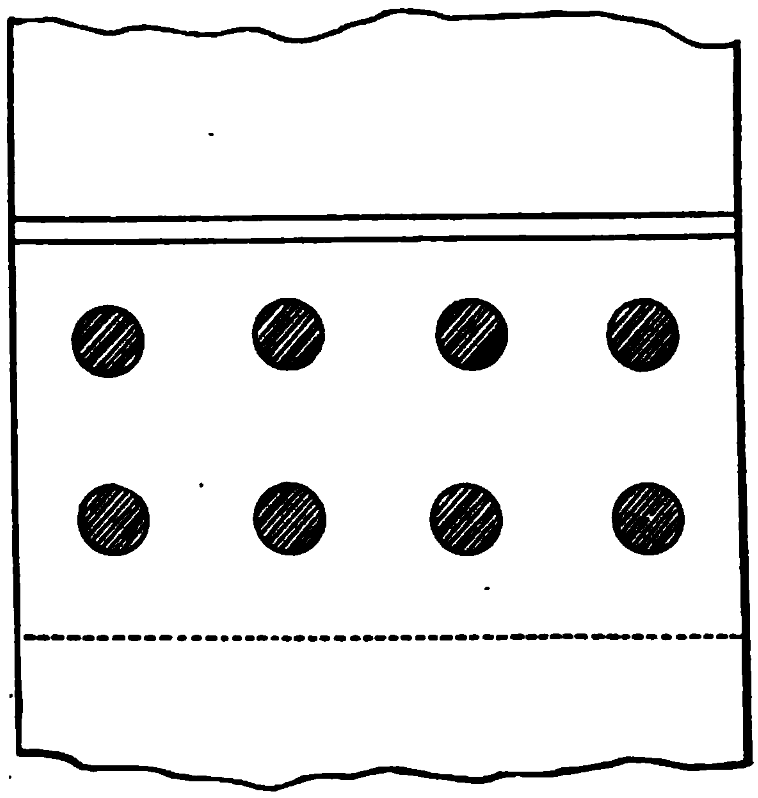

FIG. 52.

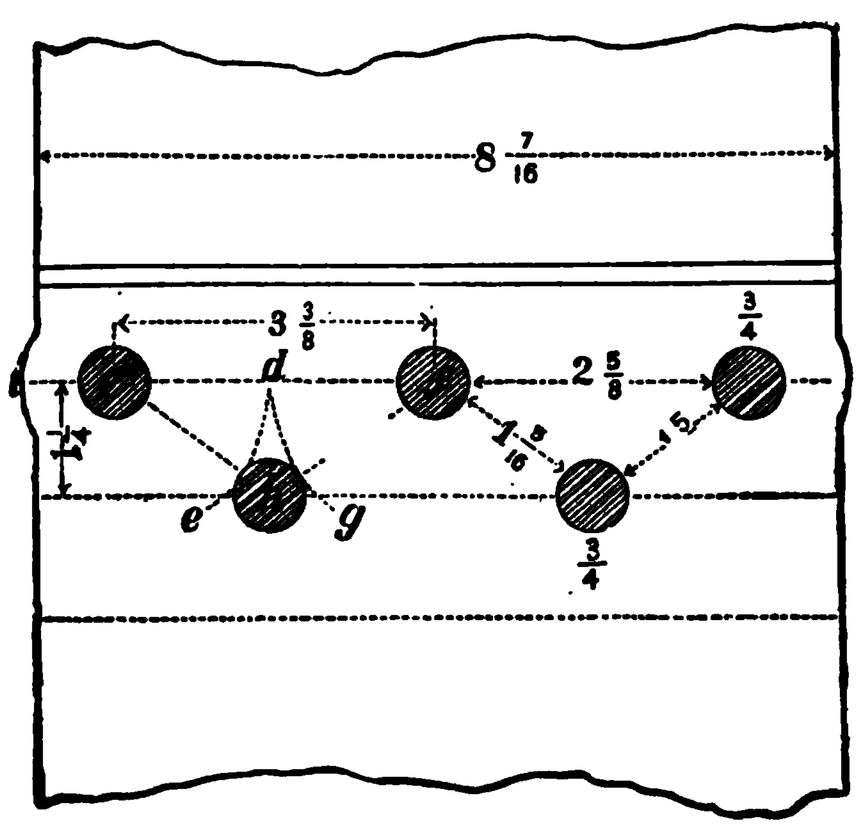

FIG. 53.

CHAPTER XVIII.

MODERN MARINE BOILERS.

*Construction.**

Cylindrical Boiler.—To avoid altogether the necessity for vertical stays as well as the transverse horizontal ones, the shell is of necessity a complete cylinder; and, although it was the demand for the higher pressures which became possible after the introduction of the surface condenser that brought the cylindrical boiler into general use, the form is often adopted now for pressures as low as 30 pounds. It is lighter, cheaper and easier to make than the box boiler, and quite as durable when worked under similar conditions; on the other hand, it occupies more space, and has not so much steam space for the same amount of grate and heating surface as the box boiler. The oval boiler (Fig. 54), however, which is a modified form, does not waste so much space as the cylindrical boiler proper, and although somewhat more expensive than the latter, it is still far cheaper than the box boiler.

There are many varieties of cylindrical boilers in use in the mercantile marine, but they may be divided generally into two classes, viz:

* Seaton.

(1) *Single-ended*, or single fired boiler.

(2) *Double-ended*, or double fired boilers.

The Single-ended Boiler has furnaces and tube only at one end, and is constructed up to as large as 17 feet diameter and 11 feet long. The chief difficulty in designing such large boilers on this plan, is to provide adequate grate area for the total heating surface which can be obtained. The number of furnaces cannot well exceed four, and is more generally three in large single-ended boilers.

Small boilers have usually two furnaces, and with this number are more efficient than with three even when of moderate size.

The number and size of the furnace must, however, depend on the size of the boiler and the heating surface it is to contain. It is found in practice that large furnaces are more efficient as coal consumers than small ones, and the reason is not far to seek. The grate area with the same length of fire-bar increases as the diameter, while the section through which the air passes, both above and below the bars, increases as the square of the diameter; it is also possible to give a good inclination or rake to the bars with a large furnace, which very materially assists combustion. In practice the fire-bars are not of course always of the same length, but they do not increase in length as the furnace does in diameter, and consequently the air passages in-

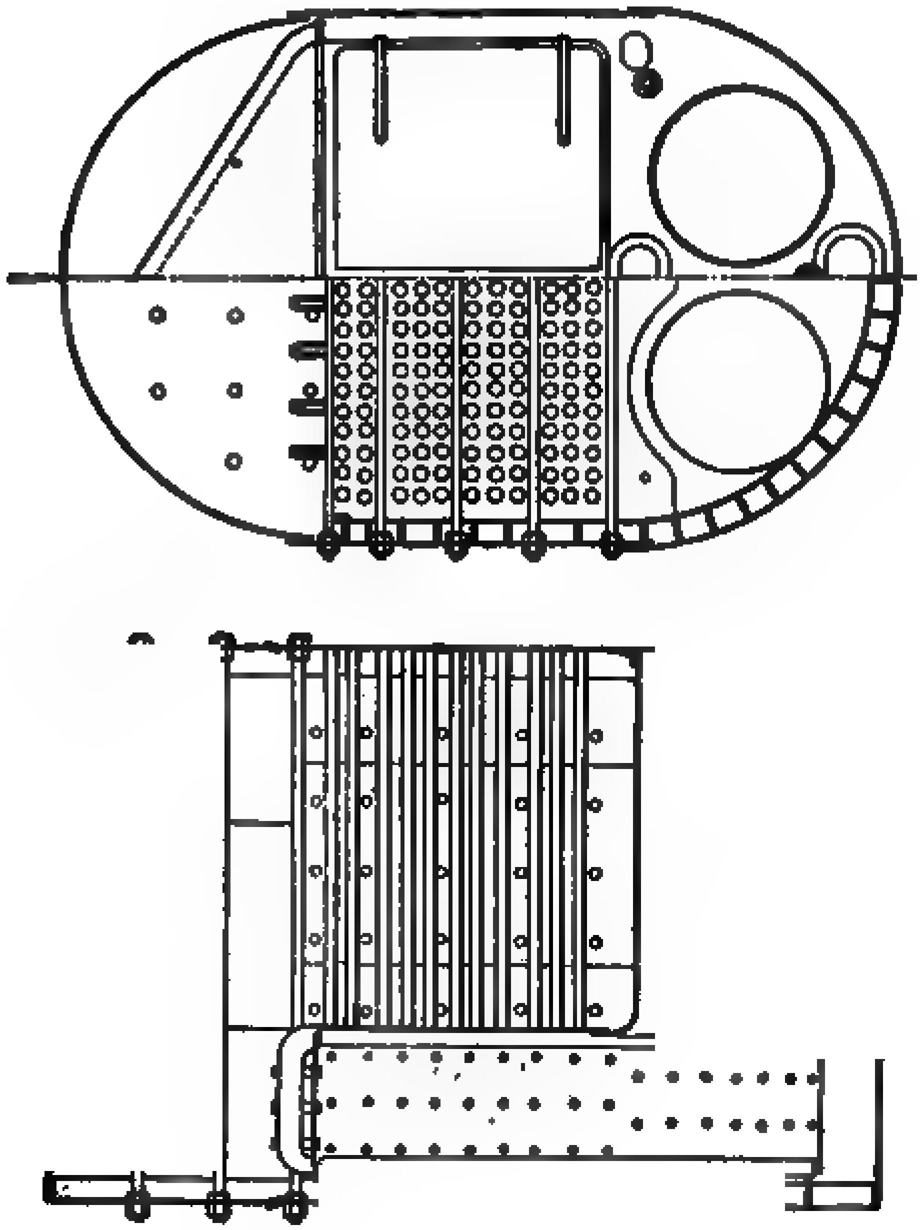

FIG. 54.—OVAL BOILER.

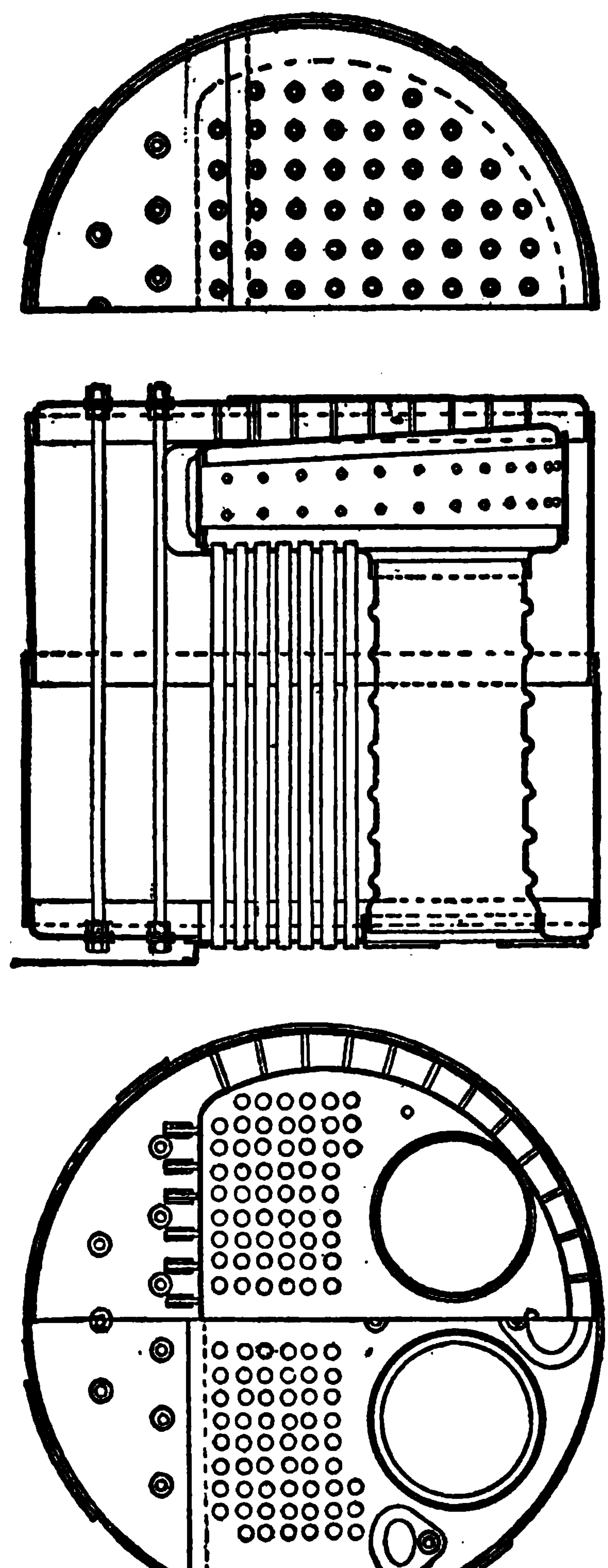

FIG. 55.—SINGLE-ENDED CYLINDRICAL BOILER (PURVES FURNACES).

crease more rapidly than does the grate area when the diameter of furnace is increased. Furnaces should not be less than 36 inches, nor more than 48 inches in diameter, except under exceptional circumstances. Taking this as a rule for guidance, boilers may be made up to 9 feet diameter with one furnace, up to 13 feet 6 inches diameter with two furnaces, up to 15 feet with three furnaces, and beyond that diameter four furnaces are necessary to avoid too great length of grate.

A single furnace boiler has of course one combustion chamber, a two-furnace boiler may have one chamber common to the two furnaces, or a separate one to each. When there is only one boiler in the ship the latter plan is preferable, as then the bursting of a tube cannot wholly disable the boiler; when there are two or more boilers one chamber common to the two furnaces is preferable, as by stoking the fires alternately an even supply of steam is kept up and the smoke consumed. A three-furnace boiler has usually three separate combustion chambers, and the same remark applies to it as to the two-furnace boiler. The four-furnace boiler has generally only two combustion chambers, one wing and one middle furnace having a common chamber; but some engineers prefer three chambers, the two middle furnaces having one in common, and each wing furnace a separate one.

The chief objection to two large furnaces instead of three smaller, and to three larger ones instead of four smaller, is the longer grate required to get the requisite area, and to the large amount of dead water between the furnaces at the bottom. There is also to be considered the limit placed by the rules to avoid risk of collapsing by direct crushing of the metal, which often prevents the adoption of the larger furnace with the higher pressures.

It is unusual and certainly most difficult to use plates above 1¼ inch thick in the construction of a boiler shell, and it is this consideration which fixes the limit of diameter. For this reason when a working pressure of 100 pounds and upwards was required, the large single-ended boiler made of iron could not be employed; indeed, 80 pounds was then taken as the limit of pressure for the very large diameter boiler.

The Double-ended Boiler has furnaces at both ends with return tubes over them, and is generally tantamount to two single-ended boilers back to back, but with the backs removed. It is made up to 16 feet diameter and as much as 20 feet long; but such very large boilers are unusual, partly owing to the want of facilities for moving such a great weight, and partly because the conditions under which such large boilers are possible are limited to very large steamers.

The double-ended boiler is lighter and cheaper in proportion to the total heating surface than a single-ended boiler, and its evaporative efficiency in practice is generally higher. On the other hand, greater care is necessary in designing and in working it. That it is lighter is obvious, and that it is cheaper may be inferred from the fact that there is less material, and less labor consequent on the reduced quantity of material.

The simplest form of this kind of boiler is one in which all the furnaces open into one common combustion chamber; this form, although at one time common enough, is now seldom adopted. The objections to it are, that the bursting of one tube will disable the whole, that the cleaning of one fire causes the efficiency to sink very low on account of the whole being affected by the inrush of cold air, and that unless special means be provided to promote proper circulation, there is a strong tendency to *prime.*

The next simplest form is one in which opposite furnaces have a combustion chamber in common, that is, it differs from the first by having the combustion chamber divided longitudinally by water spaces. This avoids the chief objections raised against the first form, while retaining its chief advantages, which are, simplicity of construction, by avoiding the flat back of the combustion chambers, with the necessary

stays, etc., and the greatest heating surface within the smallest limits of length. It is often urged against this form of boiler that the tubes are very liable to leakage at their back ends, arising from the rush of cold air against the tube plate when the door of the furnace opposite it is open, causing it to buckle. It sometimes happens that the tubes in this kind of boiler do show a tendency to leak, but it is then generally due to the want of expansion on the part of the first row of stays above the combustion chamber, when they are placed too close to the tubes. If these stays are at least 12 inches above the tubes, so as not to hold the front tube plates too rigidly, then when steam is being got up the expansion of the tubes simply causes the plates to spring very slightly, instead of to start their ends and cause them to leak. The leakage from springing of the tube plate from exposure to cold air can only take place when the combustion chamber is unduly short, and when there is an insufficient number of stays to the tube plates.

This particular form of boiler is very generally used; the evaporative results obtained from it are most satisfactory, and experience does not show it to be liable to more leakage than other boilers. Common care only is required in raising steam, and the opening of fire-doors to check evaporation is a reprehensible practice at all times and for all boilers. A brick semi-partition

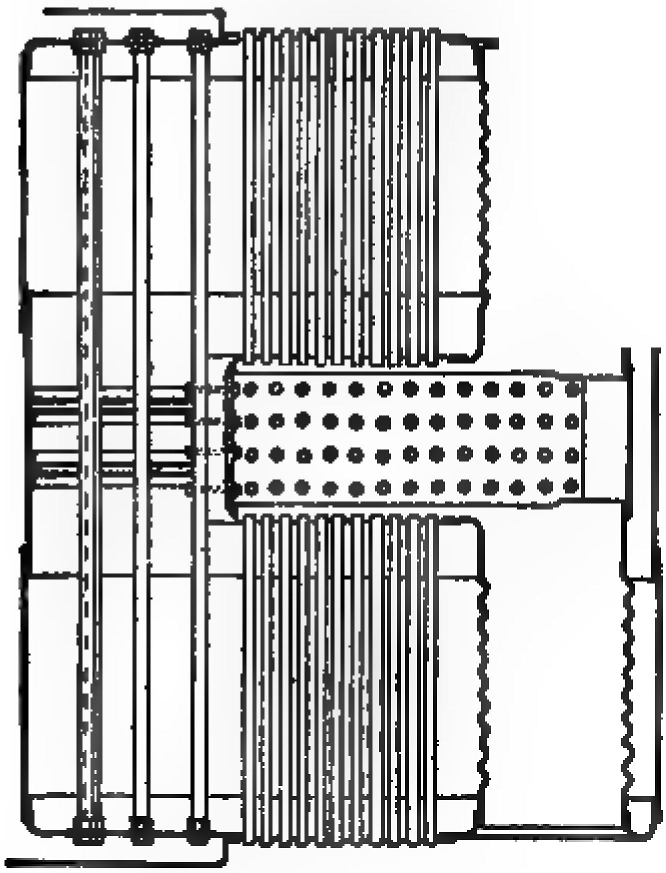

Fig. 56.—Double-ended Boiler (Fox's Furnaces).

in the middle of the combustion chamber will prevent the cold air rushing on to the opposite tube plate, and it acts also as an equalizer of temperature in the combustion chamber at all times. If, however, the combustion chamber is too small, this will only magnify the defect by causing intense local heat, and thus tending to crack the plates.

Another form of double-ended boiler (Fig. 56) has the furnaces at one end with one chamber common to them, and those at the other end with another chamber in common. The boiler is then longer than either of the other forms, and more expensive ; the combustion chambers have large flat backs, requiring a very large number of stays, which prevent their being properly cleaned from scale.

The last form, which is by far the most expensive and heaviest, but is still often adopted, is one in which each furnace has an independent combustion chamber. There is little need of description, as it is to all intents and purposes as two single boilers, except that the water and steam are common to the two parts.

Oval Boilers are included under the generic term of cylindrical, as they partake of the principal features of that class. The transverse section is, however, not an ellipse, but is really formed by two semicircles with a rectangle intervening between them. The flat sides thus

left between the semi-cylinders require staying, the first rows being at the commencement of the flat. There are both single- and double-ended oval boilers, which for pressures under 80 lbs. may be made both simply and economically to very large sizes, as the thickness of shell plate depends on the diameter of the cylindrical part. Two very large furnaces may be thus fitted into a cylindrical part of comparatively small diameter, sufficient heating surface being obtained by giving the requisite height. This form is most convenient when the boilers have to be stowed fore and aft, when the diameter is limited by the breadth of the ship between the stringers.

Dimensions of a Boiler.—The amount of grate area is the consideration which chiefly affects the choice of dimensions of boiler, and to a very large extent the number and form of the boilers also are governed by it. The rectangular boiler can be made of any breadth without in any way affecting its length or height, so that the number of furnaces can be settled arbitrarily, and any addition to the number only means some additional breadth. In small ships with the boiler athwartships and *fore and aft* stoking, the breadth of the ship does place a limit to the breadth of boiler, even when rectangular, but it seldom operates so as to seriously interfere with the boiler arrangement. The cylindrical

boiler is not so elastic in the hands of the designer; to increase the number of furnaces in it the diameter must be increased, which means that both breadth and height are affected. If two furnaces of 40 inches diameter be the limit for a boiler 10 feet in diameter, that there may be adequate heating surface, and 14 feet is the suitable diameter for three furnaces of 40 inches diameter, the grate bars being of the same length in both cases, the increase in boiler capacity is 96 per cent. for an increase of 50 per cent. of grate. The *smallest* diameter of shell into which three 40-inch furnaces can be fitted so as to give adequate heating surface, is 13 feet 6 inches, which is an increase in capacity of 82 per cent. over the boiler 10 feet in diameter. Four 40-inch furnaces require a shell of at least 16 feet diameter, which means an increase of 156 per cent. to obtain 100 per cent. increase of grate. To arrange four 40-inch furnaces so as to be convenient for stoking, a shell of 17 feet diameter is required, which means an increase of 189 per cent. over the shell of 10 feet diameter; if, instead of increasing the number of furnaces by increasing the diameter of shell, the number of shells be increased, the space occupied is considerably in excess of the direct ratio of grate areas.

It is true that to some extent increase of grate area may be obtained by increasing the length

of furnace, but the *efficiency* of a grate in practice is nearly inversely as its length; for a long grate cannot be nearly so well attended to as a short one, nor is the air supply either under or over the bars so good with a long furnace; since the area of section at the mouth, with the same diameter of furnace, is the same whether the bars be short or long.

Area of Fire Grate.—The area of fire grate required for the evaporation of a certain weight of steam depends on the quantity and quality of the fuel burned on it; the quantity of coal is generally dependent to a large extent on the quality. It may be assumed that one pound of good steam coal will evaporate 10 pounds of water in the ordinary marine boiler, 7 pounds in a locomotive boiler, as fitted to torpedo boats, when not being forced, and 6 pounds when forced to the utmost; also that in the mercantile marine, where the coal is only of average quality, 8 to 9 pounds is a fair result, and 6 to 8 pounds only can be obtained with the coal supplied in some foreign ports. The quantity of coal burnt on a square foot of grate per hour with natural draught is about 20 pounds, under favorable circumstances; with good stoking and very good draught as much as 25 pounds may be consumed; but under ordinary circumstances only 15 pounds should be supplied to obtain complete combustion and economical results.

From this it will be seen, (1) that the greatest weight of steam evaporated per square foot of grate per hour, under the most favorable circumstances, is 10x25, or 250 pounds; (2) that with bad fuel and economical stoking it may be only 6x15, or 90 pounds; (3) that with fairly good fuel and favorable circumstances it may be 9x20, or 180 pounds, and (4) that with fairly good coal and careful stoking about 150 pounds may be expected. In practice, therefore, for trial trips with choice coal and picked stokers, calculations may be based on an evaporation of 250 pounds; for mail steamships using good coal, calculations should be based on an evaporation of 150 pounds; and if a ship is going to trade in the East or localities where inferior coal is to be used, the boilers should be designed on the assumption of an evaporation of only 100 pounds of water per square foot of grate.

If the weight of steam required per hour for a given engine be calculated, and divided by one of these numbers, the result will be the number of square feet required.

If the draught be increased by artificial means, the quantity of fuel consumed per square foot of grate may be as high as 100 pounds per hour, with an air pressure of 6 inches in the stokehole; and 50 pounds with only 2 inches, the corresponding evaporations being 570 pounds and 350 pounds per square foot of grate.

The consumption of fuel per I. H. P. per hour for engines working at full power is 4 pounds, with surface-condensing expansive engines, using steam of 30 pounds pressure above the atmosphere; 3¼ to 3½ pounds with similar engines of best make and large size; 2¾ pounds with compound naval engines when forced, and 2¼ to 2½ pounds when of moderate size and working at two-thirds power; 2¼ pounds with compound engines of moderate size and as generally fitted in the mercantile marine when working at full speed; 2 pounds with mercantile compound engines well designed and carefully worked at sea full speed; 1¾ pounds with large compound engines as fitted in modern mail steamers when working at sea full speed under favorable circumstances; 1½ pounds with good triple expansion engines using coal of good quality, and 1⅔ pounds when ordinary steam coal is used; the consumption of water with these engines being about 12½ lbs.; the consumption in torpedo boats is 3½ to 4 pounds when working nearly full speed.

Assuming the consumption of coal to be 1½ pounds per I. H. P. per hour, and the grate to burn 15 pounds per square foot, there should be 0.1 square foot of grate per I. H. P. If the sea full speed I. H. P. of a merchant ship be multiplied by 0.1, the result is the grate area required for that power.

On trial trips with good coal and good stoking, and the engines working at full speed, the triple compound engine will develop 14 I. H. P. per square foot of grate; hence, one-fourteenth of a square foot per I. H. P. may be taken as the proper allowance in designing furnaces for engines to develop a certain power on a trial trip or other favorable occasion.

As the sea full speed power is usually, on long voyages, about three-fourths that developed on a trial trip, the proportion of $\frac{4}{3} \times \frac{1}{14}$ or 0.095 of a square foot per I. H. P. developed at sea,* corresponds with that given above.

Heating Surface.—Strictly speaking, all surfaces exposed to heat which are capable of absorbing, and their bodies of transmitting that heat to the water or steam, are heating surfaces; but technically only certain parts are reckoned as *effective* heating surface, and the aggregate of such surfaces is called the *total heating surface.* The surface of the upper half of the furnace, or the part above the level of the fire-bars, that of the combustion chamber above the level of the bridges, and back plates, including the actual surface of the back-tube plates, are reckoned as the effective heating surface of furnaces and chambers, and are stated separately, chiefly on account of the metal forming them

* With good triple-expansion engines 0.08 of a square foot per I. H. P. is sufficient.

being three or four times the thickness of the tubes. The surface of the tubes measured externally—that is, the area obtained by multiplying the external circumference by the length *between* the tube plates—is called the *tube surface.*

The *front* tube plates should be, and usually are, omitted in calculating the total heating surface, as they cannot be considered as effective.

The *amount* of total heating surface must depend on the quantity and quality of the fuel burnt on the grates in a fixed time, and also on the quality of the surface, &c. But since a grate may at some time have to burn the best of fuel, the total heating surface should be adequate for such an occasion.

Tube Surface.—When possible, there should be 1.0 square foot of brass tube, and 1.33 square feet of iron tube for each pound of coal burnt per hour; that is, in the ordinary marine boiler there should be about 20 square feet of brass tubes, and about 27 square feet of iron tubes per square foot of grate.

Since on trial trip with compound engines 10 I. H. P. are usually developed per square foot of grate, there should be 2 square feet of brass, and 2.7 square feet of iron tubes per I. H. P. developed on trial trip, or 2.66 and 3.6 square feet respectively per I. H. P. developed at sea.

And since with triple-expansion engines as much as 14 I. H. P. are developed from a square foot of grate, there need only be 1.93 square feet of iron tube surface per I. H. P. in the boilers for them.

Total Heating Surface.—The ratio of tube surface to the total heating surface is about 0.8 in the single-ended, and from 0.83 to 0.88 in the double-ended cylindrical boilers; taking the average at 0.84, the following will be the allowance of total heating surface based on the above consideration.

	Tubes	
	Brass	Iron.
Total heating surface per pound of coal	1.19	1.58
" " " square foot of grate	23.8	32.
" " " I.H.P. trial trip (compound). .	2.38	3.2
" " " " at sea " . .	3.16	4.27
" " " " trial (Admiralty) (com.)	2.78	
" " " " " (Mercantile) "	3.0	3 5
" " " " " " (triple)	. . .	2.3
" " " " at sea " "	. . }	3. to 3.5

The total heating surface in the locomotive boiler of torpedo boats should be based on considerations similar to the foregoing; but as weight of machinery is of more consequence than economy of fuel, and as economy of fuel can be effected by working at reduced speeds, such as would be necessary from other considerations when making long runs, the heating surface is not generally so large as would be thus given. For example, a torpedo boat burns

nearly 100 lbs. of coal per square foot of grate per hour when running at full speed with a *plenum* of 6 inches; by the rule given for ordinary boilers, there should be 119 square feet of heating surface per square foot of grate; in practice there are only 34 square feet. When the *plenum* is only 2 inches, about 50 lbs. of coal are consumed per square foot of grate, and although the above allowance of heating surface is small for this quantity of coal burnt, it is more in accordance with what is necessary for economical evaporation, and experiments have shown that the evaporative efficiency is then nearly 20 per cent. higher than at full speed. The locomotive boiler under these circumstances is a rapid generator of steam, if not an economical one; for, with a *plenum* of 6 inches, 18 pounds of water are evaporated per square foot of heating surface, and nearly 11 pounds with the 2 inches. The modern locomotive boiler on railways has usually from 60 to 90 square feet of heating surface per square foot of grate.

Internal Pipes should be fitted from the stop-valves to the highest part of the boiler, and be made with holes or slits, whose collective area is equal to twice the area of section of the pipe.

The chief object of this pipe is to collect the steam gently from every part of the boiler, so as to avoid setting up a strong current in one particular direction, and thereby induce priming.

These pipes are usually made of brass, but some engineers prefer copper, and others make them of cast iron to avoid risk of galvanic action and reduce the cost. By fitting an internal pipe, the stop-valve can be placed in a position convenient for examination and working, and it should always be so situated as to be easy of access at all times. Arrangements should also be made for opening and shutting it without going into a position of danger or difficulty, and this can always be effected by lengthening the spindles, or fitting chain gear.

In the mercantile marine, the stop and safety-valve boxes are almost invariably made of cast iron; the valves, seats and spindles being of gun-metal.

Check-valves.—Each boiler should be fitted with a self-acting non-return valve, through which the feed-water is pumped. It should also have a screw spindle, which may be used to regulate the lift, or to shut it down when water is not required. There should also be a similar valve through which the donkey pump can discharge water to the boiler.

The valve is generally of mushroom form, and made similar to the ordinary stop-valve, except that it is detached from the spindle. It is made wholly of gun-metal, and should be very strong, as at times the pressure on it may be excessive.

There should be 6 square inches of clear area

through the valve and pipe for every hundred pounds of water evaporated; or, put in a more convenient form,

Area through main feed-valve in square inches

$$= \text{total heating surface in square feet} \div 240;$$

and area through donkey feed-valve in square inches

$$= \text{total heating surface in square feet} \div 300.$$

As the feed-valves cannot always be placed on that part of the boiler best suited to receive the feed-water, and also in order to distribute that water so as to avoid its affecting the boiler plates, and internal pipe should be always fitted. To avoid the necessity of blowing the boiler down in case of accident to the feed-valves, it is a very common practice to fit these valves high up on the boiler, even in many cases above the water-level. This plan also has the advantage of providing a means of warming the feed-water, than which nothing is more essential for the preservation of the boiler; the heating is effected by the passage of the water through a long internal pipe of brass or copper, which leads it to where there is a down current of water, so that the comparatively cold feed-water may not interfere with the circulation.

Some engineers prefer to inject the feed-water in the form of spray, either above or a little way

beneath the surface of the water in the boiler; this avoids all chance of injury to the boiler plates, as any gaseous matter mechanically mixed with the feed-water is at once given up and mixes with the steam.

Great care should be taken in any case that the internal feed-pipes "run full;" that is, that they are never filled with steam, but always with water.

The *dynamic* effect of the steam in the feed-water, when mixed inside the pipe, is very startling; every stroke of the feed-pump produces an explosion, and in a very short time both external and *internal* pipes are damaged seriously.

To avoid this, the internal pipe should, when discharging above the water-level, be *turned upward* at the end, so as to always remain filled with water; and when turned downward to discharge under water, the end should be well below the lowest working level.

An additional means of safety is sometimes afforded by fitting inside the boiler a clack valve, so arranged as to close over the end of the internal pipe or on the spigot of the ordinary check valve; when this is provided, the latter can be examined when steam is up. A cock is also sometimes fitted close to the check valve, so that the supply can be regulated by it, instead of by interfering with the lift of the check valve.

Blow-off Cock.—A cock should be fitted at or

near the bottom of the boiler, to answer the double purpose of admitting sea-water before getting up steam, and to *blow off* some of the water when required. This cock should be a very strong one, as it is liable to rough usage, and being out of sight and not easily got at, it is very apt to be neglected. For this reason, as well as because a large cock is difficult to open and shut, some engineers prefer a valve to a cock. If a cock is fitted, it should be so arranged that its handle or spanner cannot be removed when it is open.

The clear area through a blow-off cock should be = 1 square inch + 0.2 square inch for each ton of water in the boiler.

FUNNEL OR SMOKE STACK.

Funnel.—This is usually of circular section, but sometimes, to minimize the transverse size of the boiler hatch, it is made of oval section. The funnels of men-of-war are often made of oval section for the same reason, but instead of the section being an ellipse, as is generally the case in the mercantile marine, it is like that of an oval boiler.

The best height to look well is four to five diameters above the taffrail, the latter when there are high bridges or boats in wake of the funnel. For the same reason, the ring for the shrouds should be $\frac{9}{10}$ the diameter from the top.

Naval ships generally have funnels with a sectional area equal to *one-eighth the area of the grate.* In the mercantile marine a somewhat larger funnel usually obtains, the area being from one-fourth to one-sixth that of the grate; in general practice a funnel, whose sectional area is one-fifth to one-sixth that of the grate, and whose top is at least 40 feet from the level of the grate, will give a very good result. The objections to a large funnel, beyond that of space occupied and cost, are resistance to the wind and large surface exposed to the cooling action of both wind and water, whereby the hot column within is partially cooled, and the draught thereby checked. On the other hand, a small funnel is liable to become excessively hot, and when the fires are freshly charged to become choked with smoke, and at all times it tends to check the draught. The funnel of a war-ship may be small, because it is so seldom that the boilers are urged to the utmost, and it must be as small as possible for obvious reasons. When the draught is forced either by a blast, or by other artificial means, the funnel may be short, and of comparatively small diameter. The area at the base of a locomotive boiler is seldom more than one-tenth the area of fire grate, and often as small as one-twelfth.

CORRUGATED STEEL BOILER FURNACES.

The use of corrugated cylindrical furnaces for internally fired boilers, that is, for boilers in which the pressure is upon the outside of the furnace, has resulted directly from the use of very high pressure steam as a necessity; the pressure now carried in marine and other internally fired boilers being impossible without the employment of these furnaces; and this pressure is continually on the increase by reason of the greater fuel economy realized from its use, and the less weight and bulk of the steam machinery using it for the development of equal power.

With the pressure of present practice, and in a still higher degree with the much greater pressure that will be needed in the immediate future, any other form of furnace is out of the question. Safety, convenience, economy in first cost and after repairs, economy of fuel, ease and rapidity of management and cleaning, all the financial, engineering and practical requirements combine to make the cylindrical corrugated furnace the only one that can be adopted in the construction of the internally fired boilers.

In fact, at the present moment, no other kind of furnace is used, unless in some cases of low pressure, which has now become the rare exception. Even in such case, however, a corrugated cylindrical furnace is both cheaper in the

first instance, and vastly more advantageous in all respects, than any plain cylindrical furnace, by reason of its greatly less weight in proportion to its resistance of a given compressive strain, by reason of its greater economy of fuel due to the nearer approach to perfect combustion which it causes, and by reason of its greater facility in handling, cleaning, etc. The weight for equal resistance to the same external pressure, of the corrugated cylindrical furnace is not one-fourth of the weight of a plain cylindrical furnace of the same diameter and length.

It should be remembered that twenty years ago steam pressure in marine practice reached 30 and 40 pounds to the square inch, being considered high at the latter figure; while, with the triple and quadruple expansion engines, pressure of from 160 to 180 pounds are demanded.

Montgomery's claim in patenting his idea of corrugated cylinders was primarily for the structural strength combined with material lightness resultant. We are not certain whether or not he anticipated or seriously appreciated the more perfect combustion that his corrugations would assure. Continued experience has demonstrated the scientific soundness of his assumptions as to strength, the processes which are used neccessitating the best quality of metal, the operation giving to the finished work a perfectly cylindrical shaping impracticable in any other mode of

production as a rule, and the pressure of corrugation not only effecting a perfect uniformity of all parts, preserving an exact standard of thickness, and perfectly graduating the angles of contour, but subjecting the material to a test which would infallibly discover any molecular defect. The natural consequence of all this is that the furnace, formerly the weakest and most sensitive feature of the boiler, has become the strongest, thereby contributing greatly to the durable character of the boiler.

The better combustion obtainable with the corrugated furnaces is a direct result of the corrugations. They thoroughly mix mechanically the current of the gases of combustion which rise from the fuel on the grate in an unmixed state, the combustible gases being in separate masses or streaks side by side with the air whose oxygen is needed for their combustion. The entire mass of gas and air when at its highest temperature should be completely mixed for perfect combustion, and the corrugations act to that effect as "riffles" on the gaseous current passing across them, so that at the upper time and place the mixing of the fuel gases with the air is mechanically accomplished with mechanical certainty. If the mixing be performed by any apparatus—special or otherwise—beyond the furnace, when the temperature of the gases will have unavoidably fallen, the economic effect will

be comparatively slight; it must be done in the furnace for maximum results, and no means have been devised so simple and efficacious as the corrugations, which perform it merely as a consequence of their form and location.

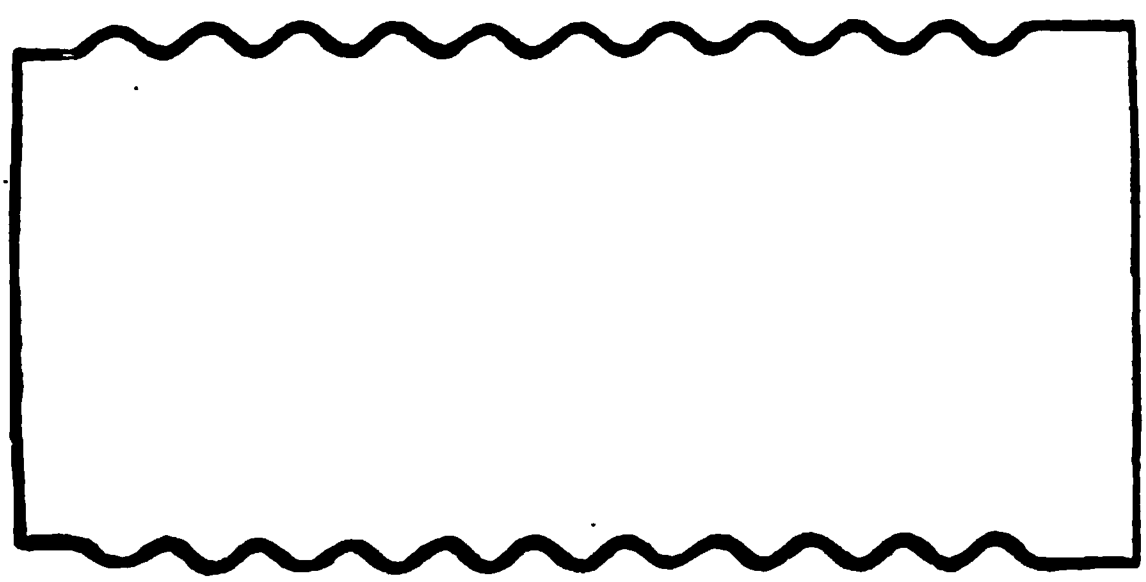

FIG. 57.—T. F. ROWLAND & CO.'S FURNACE. (AMERICAN.)

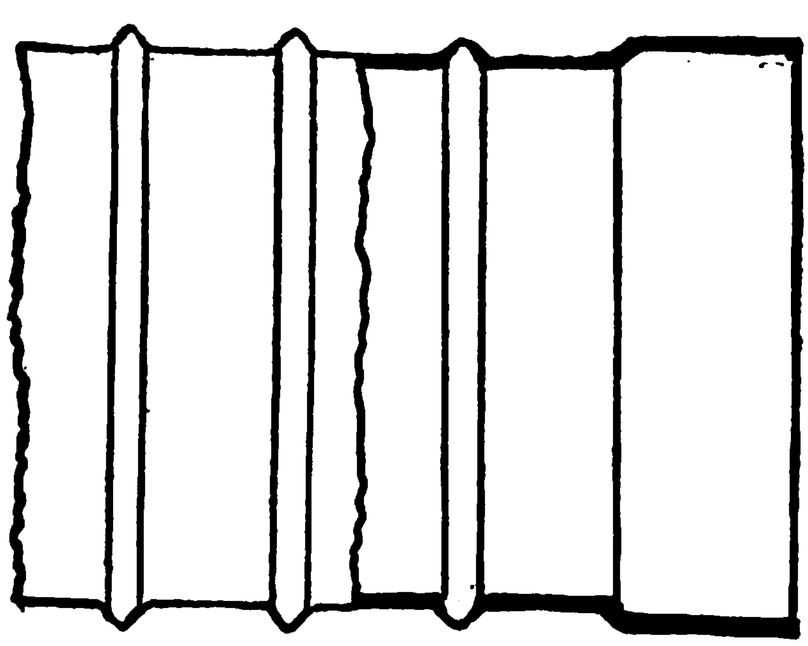

FIG. 58.—PURVES' FURNACE. (ENGLISH.)

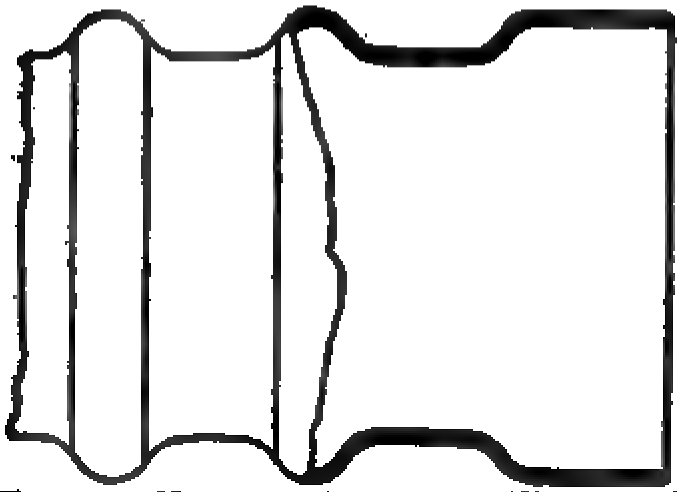

FIG. 59.—HOLMES' FURNACE. (ENGLISH.)

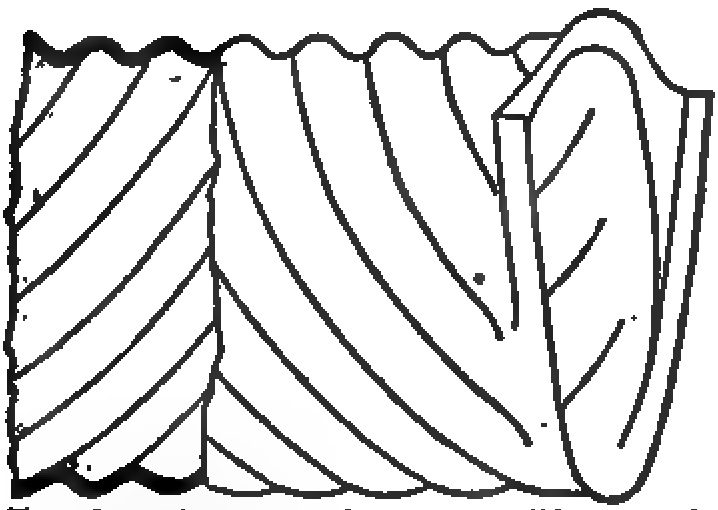

FIG. 60.—FARNLEY FURNACE. (ENGLISH.)

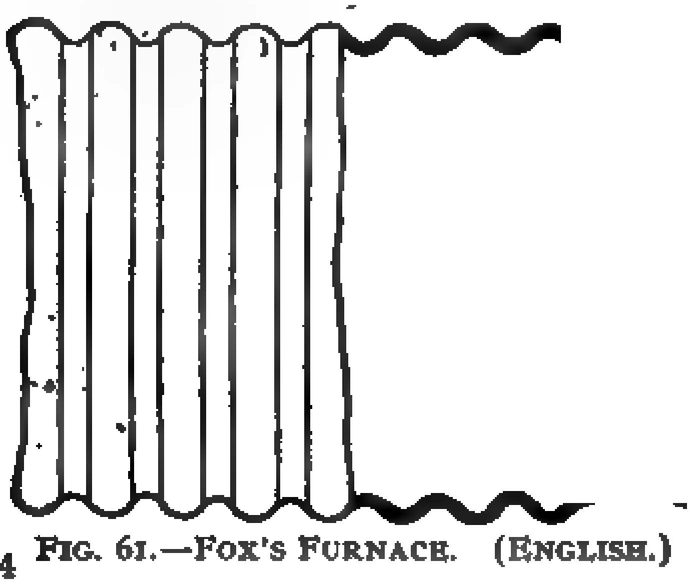

FIG. 61.—FOX'S FURNACE. (ENGLISH.)

CHAPTER XIX.

RIVETED SEAMS.

"THE pressure for any dimensions of boilers not found in the table annexed to these rules, must be ascertained by the following rule, viz: Multiply one-sixth ($\frac{1}{6}$) of the lowest tensile strength found stamped on any plate in the cylindrical shell by the thickness—expressed in inches or parts of an inch—of the thinnest plate in the same cylindrical shell, and divide by the radius or half diameter—also expressed in inches—and the sum will be the pressure allowable per square inch of surface for single riveting, to which add twenty per centum for double riveting."

So runs the beautifully simple rule prescribed by our Board of Supervising Inspectors of Steam Vessels for ascertaining the pressure allowable for dimensions of boilers.

Some two or three years ago we were engaged designing a marine boiler to have as thin a shell as possible consistent with safety for a light-draft steamer, and great was our surprise when informed that, though we made the longitudinal seams triple-riveted according to the most approved practice, we would not be allowed any

* Oldham.

reduction in thickness of shell plates below that due to an ordinary double-riveted joint, though we venture to submit that the triple-riveted butt joint, as generally obtaining in ocean steamships, is quite ten per cent. stronger than the best double-riveted joint, and is twenty per cent. stronger than the ordinary double-riveted lap

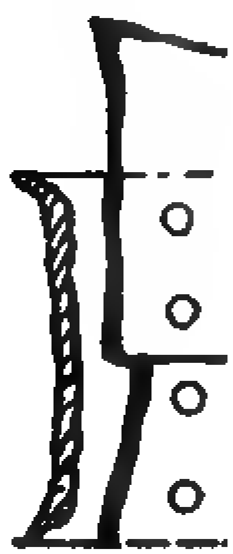

FIG. 62.—ORDINARY DOUBLE-RIVETED LAP JOINT.

joint. This means that at least fifteen per cent. in weight of shell could be saved without any sacrifice of strength.

To make this clear, let the accompanying drawings show an ordinary double-riveted lap joint, an ordinary triple-riveted double butt strap joint and the most approved triple-riveted double butt strap joint, respectively. The per-

centage of strength of joint, as compared with the solid plate, is 59 for double-riveting, 72 for ordinary triple, and 83 per cent. for the improved compound triple, as per Fig. 64. With regard to this butt, it has been objected that eight and a half inches pitch is too great for solid caulking, but in reply we may state that the correct proportions of these very dimensions of joint is

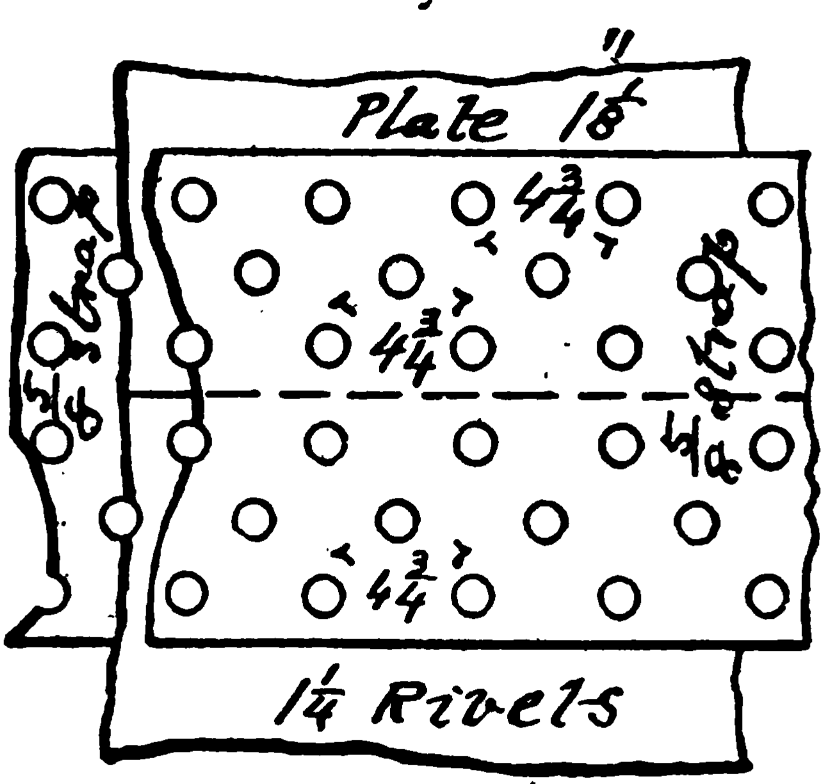

FIG. 63.—ORDINARY TRIPLE-RIVETED DOUBLE BUTT STRAP JOINT.

placed beyond the region of doubt, as many such are now afloat on the ocean, after having been tested to three hundred and twenty pounds hydraulic pressure per square inch, without showing any signs of leakage or distress, and now carry one hundred and sixty pounds constant working pressure above the atmosphere without leakage—always excepting the inevitable bottom leakage, due to expansion and contraction.

For any given plate and any given size of rivets, there is always only one proper pitch, which cannot be departed from without a sacrifice of strength. Fig. 64 and table marked A is drawn to show the possible gain by a correct system and proportion of riveting. It commences with an ordinary triple-riveted joint, in accordance with Lloyd's Register rules, and shows that the strength of plate at butt is forty-five per cent. less than the section of solid plate; an improved triple butt with three complete rows of rivets, reduces the loss to thirty-five per cent.; with a compound triple and larger rivets, the loss is further reduced to twenty per cent.; and lastly, this loss amounts to only five per cent., the strength of riveted joint being now equal to ninety-five per cent. of the solid plate.

The sheer strakes and upper stringer plates of long, large vessels should always have double straps, and the bilge strakes may be treated in the same way with advantage; for the gain in strength is great, and the loss in speed by augmented surface friction is unappreciable. We assert this after having had outside straps fitted on the bilge and bottom of many steamers, of high and low speeds, for continuation of class in the *Veritas* and Liverpool Lloyd's Registry. It is said that these straps do not *look nice* —a common expression, but one, we venture to think, which ought to be used with a greater

degree of caution by engineers, for to my mind *what is nice looks nice.*

Diagram No. 65 and table illustrate, on the left-hand side, the proportionate loss sustained by various thicknesses of plates, from three-eighths to one inch, with riveting in consonance with Lloyd's Register rules. On the other hand, it shows this loss largely reduced by adopting a proper system of riveting, whereby fifteen pounds

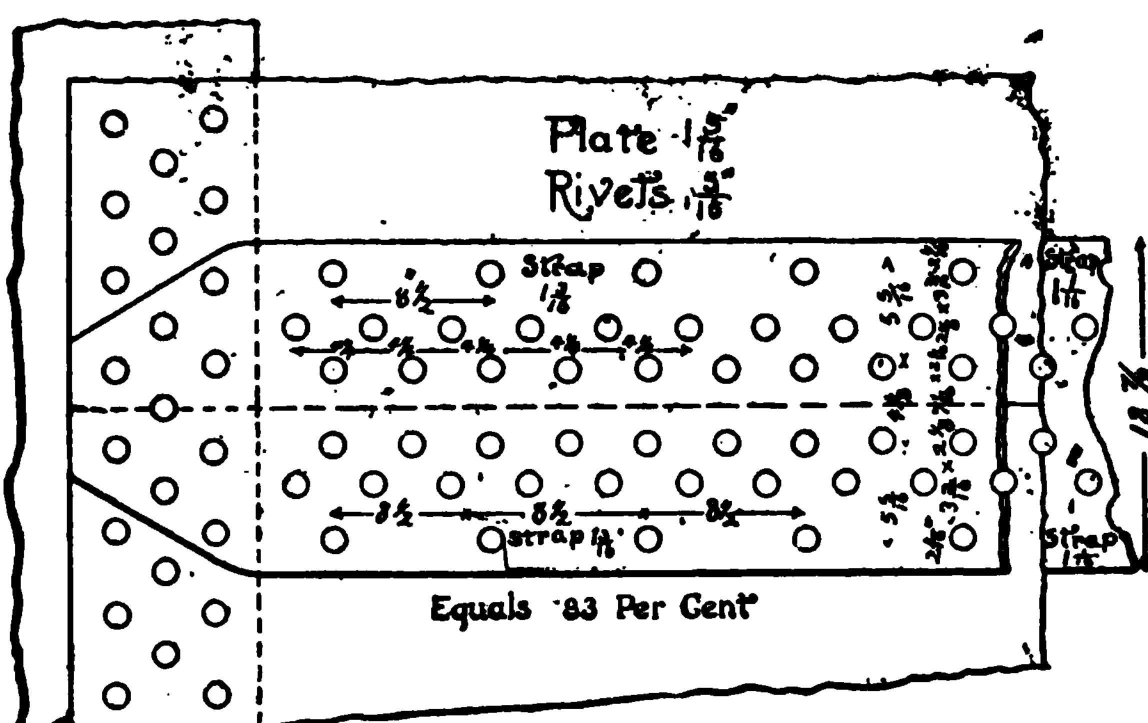

FIG. 64.—IMPROVED COMPOUND TRIPLE-RIVITED DOUBLE STRAP JOINT.

of plate are made to exceed twenty, twenty-five exceed thirty-five, and thirty-five pounds still more largely exceed forty in strength; the average gain being over 28 per cent., and that without any sacrifice of strength.

These percentages are calculated for counter-sink riveting, which, we need hardly say, is quite

a different thing to parallel holes; for, though the countersinking increases the size of rivet hole some twenty-five per cent., and thereby reduces the strength of plate for a given pitch, it adds nothing to the shearing strength of the rivets. It is, indeed, a premium that has to be paid for flush riveting.

The annexed table (B) shows the size of pitch, butt straps and description of riveting to obtain from 80 to 83 per cent. strength of joint, as compared with the solid plate for various thicknesses from three-eighths of an inch to one inch. This table shows that in riveted joints of structures composed of malleable iron or mild steel plates in which the joint is the means of the continuity of strength throughout the structure, that a reduction of about twenty-five per cent. might safely be made in an ordinary steel steamer, of, say, 2,500 tons gross register, if the arrangement of butts were good, the rivet holes in vital parts absolutely fair, and the rivet area and pitch of correct proportions.

It would further appear from this, that about 150 tons weight might be saved in steel plates of ships and boilers, worth, say, fifteen thousand dollars, and the dead weight ability of the vessel be augmented some five per cent., or with the same weight of metal the length and breadth of the vessel could be materially increased, whilst retaining the original factor of safety.

TABLE A.

SHOWING LOSS IN STRENGTH OF PLATE BY THE ORDINARY SYSTEM OF RIVETING AND GAIN BY IMPROVED SYSTEM.

Weight of Solid Plate.	REDUCTION. Net Effective Section of Plate left after Riveting.	REDUCTION. Net Effective Section of Plate left after Riveting.	Weight of Solid Plate.
Pounds.	Pounds.	Pounds.	Pounds.
40	22	29	35
35	20	20	25
30	17	18½	22½
25	14	14½	17½
20	12	12½	15

TABLE B.

PERCENTAGE OF STRENGTH OF RIVETED JOINTS.

Thickness of Plate.	Diameter of Rivets.	Pitch of Rivets.	Number of Straps.	Description of Riveting.	Percentage of Strength.
Inches.	Inches.				Per Cent.
3-8	⅝	{ 2½ / 5 }	1	Triple.	82
7-16	¾	{ 2¾ / 5½ }	1	"	82
8-16	¾	{ 2½ / 5 }	1	"	80
9-16	⅞	{ 3 / 6 }	1	"	81
10-16	⅞	{ 2.9 / 5.8 }	1	"	80
12-16	⅞	{ 3¼ / 6½ }	2	"	82.7
14-16	⅞	{ 3¼ / 7 }	2	"	83
16-16	1	{ 4 / 8 }	2	"	82

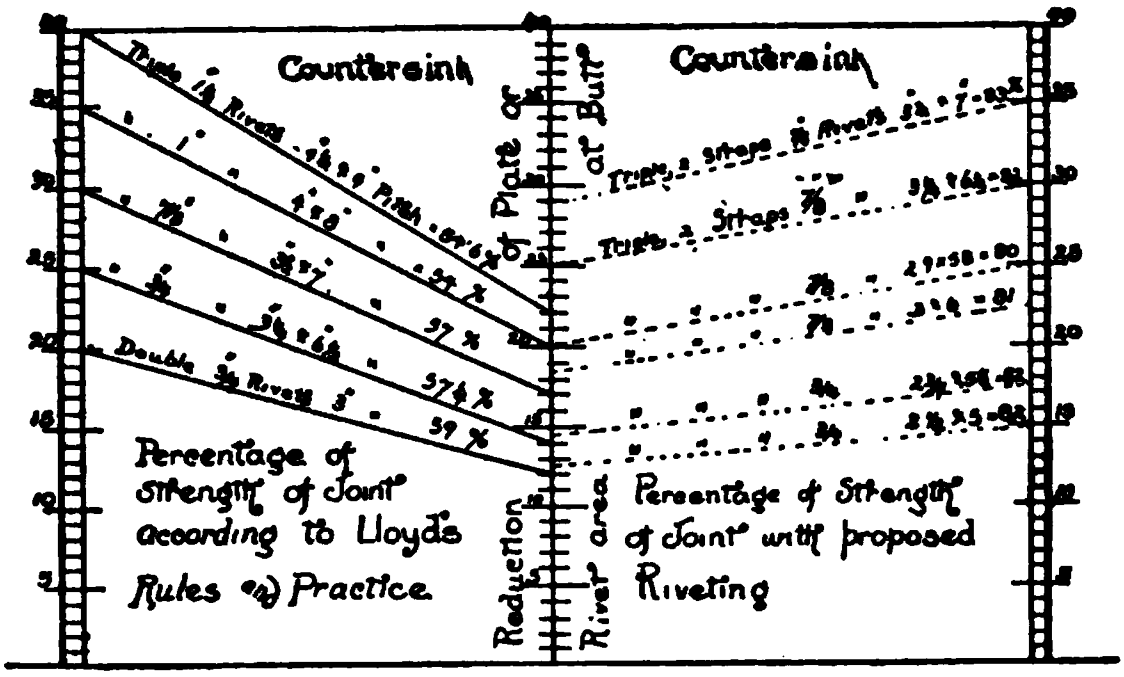
Countersink
Countersink
Percentage of Strength of Joint according to Lloyd's Rules and Practice
Percentage of Strength of Joint until proposed Riveting
Reduction of Plate at Butt
Rivet area
Double ¾ Rivets
57 %
57½ %
59 %

FIG. 65.

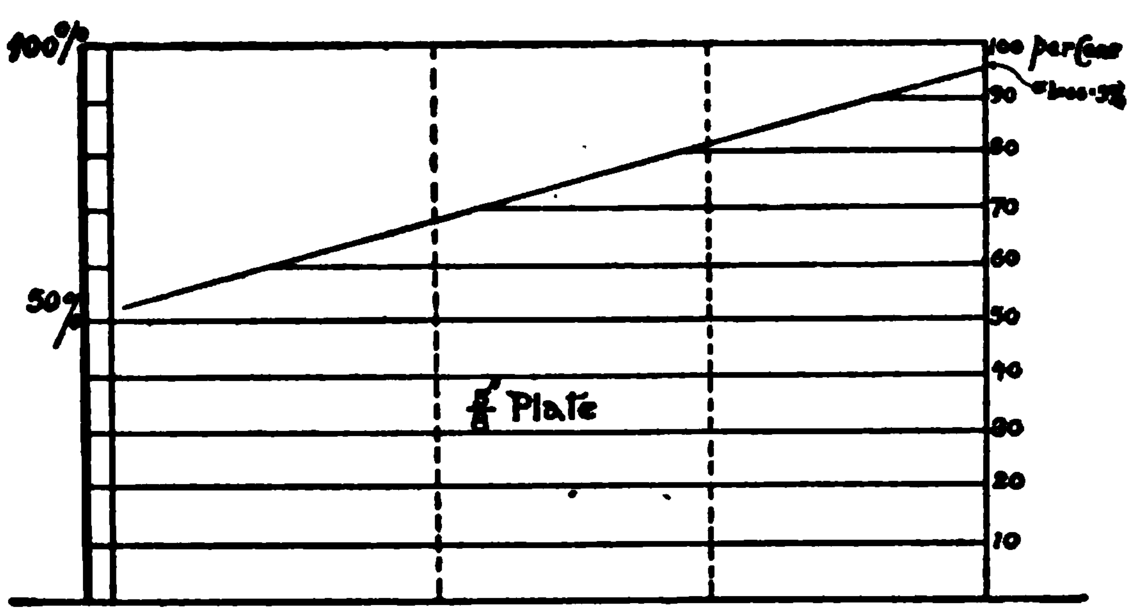
100%
50%
100 per cent
90
80
70
60
50
40
30
20
10
Plate

FIG. 66.

CHAPTER XX.

FORCED DRAFT.*

FORCED DRAFT for boilers, though little used on shore, is becoming a favorite practice in the marine service. The ordinary natural draft in a chimney is equal to a pressure measured by a column of water about $\frac{1}{4}$ inch high. Since the pressure of the air is about 14.7 lbs. to the square inch, or is measured by a column of water about 34 feet high, we see that this pressure of air in the chimney is only about $\frac{1}{100}$ of a pound on the square inch. Now forced draft increases this pressure often 20 times. This increase in furnace draft results in a rapid increase in the coal burned per square foot of grate area per hour. An increase in consumption of fuel calls for an increase in the ratio of heating surface compared to the grate surface, for otherwise the tubes would leak, or the plates would become overheated. Also, the temperature of the waste gases in the chimney is generally increased. Hence it is not usual to find that any gain in the economy of the fuel is realized with forced draft, for the water evaporated per pound of coal burned is invariably decreased.

* Whitham.

To offset these disadvantages, forced draft enables a greater boiler power to be realized than can be had with the same boilers under natural draft. In one case by applying forced draft to old boilers, four developed the same power that had been given by eight. In several cases it has been claimed that the boiler power has been increased 50 per cent. This means that in designing a new plant, one boiler with a strong forced draft will develop the power required by two with natural draft. Hence there is a decrease in the first cost of the plant; not, however, a decrease of one-half, since the forced draft boiler must have a greater heating surface than would be required in one of the two boilers using natural draft. Besides, forced draft means that some of the power developed by the boilers must be used in driving a blower. Yet the gain due to decrease in the number of boilers is great. On board ship, space and weight are items of great importance, for they mean that a greater cargo or a more powerful battery may be carried.

There are two main ways in which forced draft is used. The first method consists in locating the boilers and fire room in an air-tight compartment, and forcing air into this compartment under the pressure of say 4 or 5 inches of water. This method permits of the air entering the furnace in the usual method, and the fires may be worked as for natural draft. Of course the heat-

ing surface must be increased in ratio to the grate surface.

2d. The other method is illustrated herewith. It consists in forcing the air into the furnace, and not into the fire room. The ash-pit and furnace doors must be about air-tight. In some patents the air is warmed on its way to the furnace by causing it to take up heat from the escaping gases of the chimney. This is done by either passing it around a lot of tubes in the uptake, through which the smoke passes, or by having the current of air pass through a drum surrounding a part of the chimney. It has not as yet been proven that the gain by these latter methods is as great as is realized by heating the feed-water by the chimney gases.

The second method does not permit of coaling of the fires unless the draft is shut off. The fires may, however, be worked by any one of the forms of rocking grates illustrated in a former article. In the present illustration the Ashcroft bars are used, and each bar is turned by a socket wrench.

In any system of forced draft it is important that the ash-pit shall be kept well cleaned, for otherwise the grates will burn out quickly.

Air must be admitted *over* as well as *under* the grates, for otherwise the combustion would be chiefly at the bottom, the grates would not last so long, and there would be a loss of fuel

by gases being distilled and passing off unburned.

Perhaps one of the very best forms of forced draft arrangements is presented herewith. It was invented by John C. Kafer, of the U. S. Naval Engineers, and has been applied to several vessels with marked suceess. Fig. 67 shows the ordinary marine type of boiler, containing two

FIG. 67.—KAFER'S FORCED DRAFT SYSTEM.

internal furnaces. The ash-pit door is closed and made tight by an asbestos gasket. The furnace door is made in the usual hollow form. The inner plate of this door is perforated with a lot of small holes. The air from the blower enters at A (the damper being moved aside), and passes into the ash-pit. From thence a part goes through the grate, while the remainder

passes through holes in the dead-plate and into the front of the furnace, passing through the numerous holes shown in Fig. 67.

FIG. 68.—KAFER'S FORCED DRAFT SYSTEM.

FIG. 69.—THE ROBERT

ERTS SAFETY WATER TUBE BOILER. Opposite Page [illegible]

CHAPTER XXI.

WATER TUBE MARINE BOILERS.

FIGS. 69 and 70 illustrate the water tube boiler invented by Mr. E. E. Roberts, of New York City. The design and construction can be seen at a glance, rendering a detailed description unnecessary.

Some years ago Mr. Roberts built a yacht for his own use, as he was a lover of aquatic sports, and had previously had much experience at sea as an engineer in the regular service of the U. S. Navy. He succeeded in designing a hull and an engine to his entire satisfaction, but after trying almost all the various types of boilers in the market, found that they were either too heavy, stood too high in the boat, carried too low pressure of steam, took too long to get up steam, or required too frequent cleaning, either of the fire spaces or of the water spaces, and were generally unsatisfactory. He then designed a natural circulating water-tube boiler, which, although crude and heavy as compared with those he is now building, turned out to be a complete success, and has been in use nearly ten years without repairs. After using the boat

for about two years, he sold her to other parties, and the result was that from whatever point the boat touched at in cruising, arrived inquiries addressed to Mr. Roberts in regard to the boiler. The demand soon became so great that he was obliged to devote his entire attention to the building of these boilers; and, although the original principle has been retained, he has made many improvements in the material and method of construction, so that at the present time he has turned out about 260 of these boilers, ranging from sizes adapted to small launches up to those used in ocean-going steamers. These boilers are now all being built in duplicate as to each separate size, and the parts are interchangeable. They are all tested before leaving the works at from 300 to 400 pounds hydrostatic pressure, and also 200 pounds of steam and upwards. They are strongly patented in the United States and other countries, and are approved by the United States Inspectors for pressures ranging from 200 to 300 pounds of steam. They are very light, occupy very little space, set low in the boat, obtaining a working pressure in ten or fifteen minutes, are economical in fuel, are absolutely safe from disastrous explosion, and are specially adapted for use in connection with modern high-pressure expansion engines.

FIG. 76.—THE ROBERTS SAFETY

ETY WATER TUBE BOILER.

OPPOSITE PAGE [illegible]

FIG. 71.—THE ALMY WATEI

Water Tube Boiler.

Opposite Page [illegible]

THE ALMY WATER TUBE BOILER. (FIG. 71.)

Built by the Almy Water Tube Boiler Co., of Providence, R. I., is made of either steel or wrought iron tubes, having fittings of malleable iron of the best quality, specially for their use from their patterns. The internal arrangement is of tubes vertical on three sides, then at sufficient height tubes horizontal. This forms fire box, the grate surface of which occupies 75 per cent. of floor space of boiler, and representing about 1 square foot grate area as equal to 25 to 30 square feet of heating surface of boiler.

The great number of square feet within fire box, where all tubes exposed to radiant heat are of small diameter; the direct vertical flow of the products of combustion up between and around the horizontal tubes, the under sides of which always present a clear surface to the heat, enables this boiler to be an extra quick steaming one. Steemships, torpedo boats, yachts, or merchant steamers provided with the Almy water tube boiler, can have full head of steam from banked fire in 10 to 20 minutes.

The feed-water is received at either front or back of boiler into lower manifold, passing first through a heater provided for that purpose, separate from, but in connection with boiler proper, from whence it enters boiler at a high temperature. The flow of water in circulation is continuous in one direction. Before completing the

circuit the current of steam, containing water, enters the separator in front of boiler, which takes the form of an involute coil. Here the dry steam rises to top of separator where it is drawn off for engine power, and here, also, by the law of gravity, the water now freed from the lighter steam is returned to the circulation fr∂m the bottom of separator to lower manifolds on each side of fire box.

Repairs to this boiler are made under conditions of great convenience. A section of casing being unbolted, sections of boiler can be unscrewed, removed and replaced by any handy engineer or fireman. When necessary, any section can be blocked off at sea, with the same celerity that a tube in other boilers can be plugged.

Tube sections always obtainable at the works, or can be cut anywhere and replaced in fittings.

The sectional form of this boiler, though usually rectangular, enables a lengthening or a widening of shape, as certain spaces may require. They are built in sizes from 3 square feet grate area to 50 square feet grate area, with height in sizes from 15 to 50 square feet grate area, from 7½ feet to 9 feet; while floor space for largest size of 50 square feet grate area requires but 8 feet square. The weight of the Almy patent water tube boiler, in the larger sizes, is about 400 pounds per square foot of

grate area and from 250 to 350 pounds per square foot grate area in smaller sizes.

These boilers are designed for 250 pounds per square inch steam pressure. They have been approved by the Board of Supervising Inspectors of the United States. There are numbers of them in use to-day in passenger steamers and steam yachts, resulting in economy in weight, space and fuel. These boilers are enclosed in sheet steel or iron casing, lined where necessary with firebrick and asbestos and throughout with asbestos. They are tested to 400 pounds hydrostatic pressure, and with all furnishings are delivered complete ready to place and unite to engine.

CHAPTER XXII.

REPAIRS AT SEA AND HOW TO MAKE THEM.

BREAKDOWNS at sea will happen, owing to structural weakness, bad design or workmanship, to the most careful and able engineers; and when they do occur the engineer in charge must then show what he really is, by the clear foresight and mechanical skill which will enable him to so repair damages that the ship can make some port in safety. Breakdowns happen at the most unexpected times and in the most unexpected places—sometimes from the ignorance or carelessness of the engineer on watch, and sometimes from causes over which he has no control. Accidents do not happen as per order, and this is the reason why no special rules can be given for the repair of all kinds of breakdowns—a rule which would work admirably in some cases would not do at all in others. An engineer should always be studying his engine, and by imagining some part breaking down, he can then calmly think over a plan of repair, so that if that part should give way, he would know at once what to do. No specific rules can be given by anybody for the thousand and one cases which may arise, but a general plan of

procedure in case of the most common and serious accidents will be given to aid the engineer in his hour of trouble.

One of the prime causes of accidents to marine engines is the inherent weakness—the structural weakness—of the *ship's hull:* iron and steel hulls, as well as wooden ones, for none of them are rigid bodies. An engine may be in perfect alignment when at rest, but be really out-of-line when at work driving the ship at sea. If the hull is weak—too weak for the power put into it —it will assume a different shape when the engine is at work to what it is when at rest at anchor or lying in the dock. In such a case the engine will be a chronic heater and thumper, and no engineer, however able, will succeed in keeping it in good working order; it will prove a hopeless case beyond all engineering doctoring.

Tinkering at such an engine will only increase the evil, and is worse than useless. It is this inherent weakness of the hull structure which causes some, but not all, broken shafts, cranks, and crank pins. Thrust bearings built on a weak and yielding foundation, spring away bodily to thrust of the shaft, which comes inboard, and has a tendency, and a bad one, to close up the after crank; the shaft brasses become cap and collar bound; and soon heating and thumping begin, and a breakdown is near at hand.

As soon as the engine is stopped, everything resumes its normal condition—everything is found in line—and the engineer wonders "What is the matter with the machine, anyhow?"

Keep an eye on the shaft and thrust bearing—a slight movement can only be detected by the all-seeing eagle eye of a true engineer.

It is troubles like these added to the troubles caused by the engine *per se* (caused by bad design, poor material, and worse workmanship), which brings the modest but able engineer to the front, and sends the ignorant, boasting braggart to the rear.

If your engine needs repairing and you are able to repair it, do it. And here I will devote some valuable space to give my brothers of the marine engineering fraternity—particularly the younger members of the craft—an object lesson on the art of "letting well enough alone" in the management of marine engines, boilers, etc. It is very desirable that marine engineers should understand how to repair their engines, boilers, etc., in case of necessity—say when at sea and the ship a thousand miles away from a shop.

Now when anything happens to the machinery and it fails to perform its functions properly, a little forethought should be used before proceeding to put things to rights. Merely taking an engine apart and fitting it together again, amounts to nothing, and is no criterion of engi-

neering skill. Before putting a hand upon it, it should be studied closely to find out *just where* the trouble is and *what* causes it, then the proper remedy can be applied,—guessing at it won't do. Nine times out of ten the trouble will be found in some inconspicuous part, something that a turn of a wrench or tap of a hammer will cure. Now, my brothers, is it not better to spend a half hour in finding out the true cause of the trouble and another half hour in fixing it, than to guess and go it blind with wrench and hammer, and dismember the whole machine?

Take a case or two by way of illustration: Go aboard the steamship *Nonesuch*, when you will, and make a bee line for the engine room. You will find the engine working like a clock, and Chief Brown, neat and tidy, taking things cool and easy. Chief Brown is a thorough engineer, but he is as modest as he is able; and you will never hear him boasting of his own abilities. He is not only a skilled practical mechanic and expert engineer, but he is a student also, devoting all his leisure time to studying books and papers on steam engineering and thereby keeping himself abreast of the times.

Now go aboard the steamship *Absurd*. Behold the old machine working like a stone breaker; see Chief Jones smeared from head to foot with grease and dirt, monkey wrench and copper maul in hand, the very picture of

despair. The perspiration rolls off his face; his countenance is agitated; he is swearing a blue streak at everybody and everything. His engine is always out of order, and he keeps it so by tinkering at it day and night. The trouble with Jones is, he don't think; he is energetic enough, but his efforts lack intelligent observation. Jones is not a student, but he is a great boaster. When he hears a thump, no matter where it really is, he seizes a copper maul and goes for the stub-end of the connecting rod, remarking as he does so: "I am a practical man, I am; there is no book engineer about X. Y. Z. Jones," while the assistant looks on in admiration not unmingled with fear. In a few minutes the wail of the greaser is heard crying, "Hot fat, Hot fat, the crank-pin is on fire!" We all know Brown, and we all know Jones. They can be found in every seaport in the United States.

The greatest menace to the modern marine engine is the meddlesome, tinkering engineer, who will tinker and experiment like the boy in the fairy story: he cuts the bellows open to see where the wind comes from, and if the average marine engine could only talk it would exclaim, "All I want is to be let alone."

We will now consider a few possible mishaps, and what we would do in just such emergencies.

CAUSES OF ENGINE THUMPING AND THE REMEDIES.

The engine may thump because the shaft, crosshead, or slides are out of line; the cranks, crank-pin, or coupling-bolts may be loose; there may be, and often is, in old engines, a collar formed in the cylinder from long use and neglect, against which the piston brings up at the end of the stroke; if such be the case, the collar must be filed or chipped off. Some of the keys in the stub end or fork end of the connecting-rod may be loose; the piston packing may need setting out; the follower bolts may need tightening up; or the valve or valve gear may be out of order.

All noises are caused by lost motion or looseness somewhere, and they can be cured if the prime cause of all is not the *weak hull* of the ship. It is good policy on hearing a heavy and unusual thump or noise about your engine, to examine all the keys and caps, to see if they are tight; if they are all right, take off the cylinder-head and see if a collar has formed in the cylinder; if not, examine the piston packing, and set it out if necessary. Try the piston itself; it may be working loose on the rod. If the air-pump is worked by the main engine, examine its piston; perhaps it will be found loose; next examine the main valve, and see if it has the usual lead; also, the eccentrics and link motion. Having done

this, and all having been found in order, if the thumping still continues, you may take it for granted that the shaft, slides, or cross-head is out of line, and you must line it up as directed below. To do this, assuming that the cylinder is in its proper place, run a line through the centre of the cylinder, measuring from different points to get the line exactly in the center, and make it fast in the crank-pit; then—

TO LINE UP THE SLIDES.

Measure with a straight-edge the distance of the slides from the line at the top and the bottom, and if the measurements are not the same, liners must be used to set them true.

TO LINE UP THE CROSS-HEAD.

If the connecting-rod has a fork-end, and there is a hole in the cross-head for the piston-rod, if the line does not pass through the centre of the hole, the cross-head is not true, and the gibs must be adjusted to make it so.

TO LINE UP THE SHAFT.

To line up the shaft, bring the crank around neariy on the top centre, just so that it will touch the string, measure the distance of the forward end of the crank from the line, then turn the crank around on the bottom centre as before, and measure the distance of the crank

again. If all the measurments are the same, the shaft is in line, that is, it stands at right angles to the centre of the cylinder; but if the measurments are not the same, the shaft is out of line, and it must be raised or lowered until the measurements are the same. Example: Suppose the crank-pin be 5 inches long, and (taking an extreme case) the distance of the string is 3 inches from the forward crank and two inches from the after crank when on the top centre, and just the reverse when on the bottom centre, it shows the forward end of the shaft is down and needs raising ½ inch. This is an extreme case of course, and the measurements will seldom differ more than ¼ or ⅜ of an inch.

This method saves the trouble of taking out the crank-shaft, which would have to be done to use a T square for the purpose.

To line up the shaft of a side-wheel steamer, put the crank on the top centre, and then on the bottom centre, and measure the distance accurately between the cranks in both positions and note the difference in measurements. Now if the difference is greater when on the top than when on the bottom centre (as it generally is) it will show that the outboard end of the shaft (one or both) is down, and one or both must be raised until the measurements are the same.

To see if they are in line fore and aft, put the crank on fore and aft *half centers* and measure

the distance between the cranks as before; if the measurements are not the same, then the outboard end (one or both) must be moved fore or aft until the distance between the cranks is the same.

A side-wheel engine is placed in the centre by similar means. You mark the slides as before directed, and then nick the shaft at the spring bearing, or in fact any stationary part with which it comes in contact.

BROKEN VALVES.

In case of a triple-expansion engine, if the high-pressure cylinder valve should break, take it out entirely, and let the steam flow through the H. P. cylinder into the intermediate cylinder. This will turn the engine into a two-cylinder compound engine. Leave the valve stem in the stuffing box to prevent steam leakage. Place the link in mid-gear and leave it as it was. If there be any pumps worked from the cross-head of the disabled engine, leave the connecting-rod as it is to work the pumps; but if not, take off the stub end (crank pin end) and sling it out of the way, leaving the piston resting on bottom of cylinder.

2d. If the valve of the "intermediate" cylinder should break, proceed as before directed; the steam will flow through the I. P. Cyl. from the H. P. Cyl. to the L. P. Cyl.

3d. If the valve of the low-pressure cylinder should break, take it out as before; the steam will then exhaust from the I. P. Cyl. through it to the condenser.

ECCENTRICS AND ECCENTRIC-RODS.

If the "go-ahead" eccentric or eccentric-rod should break, take it off and substitute the "backing" eccentric or rod (as the case may be) in its place. Then sling the backing-end of the link up to prevent its sliding on the link block. Make fast to side of ship or some part of engine frame. Keep eccentric-rod in line with valve stem. In such a case, bear in mind, the engine cannot be "backed." Repair the broken part if possible, and use in place of part taken away.

DAMAGED CRANK-PINS.

Cracks or flaws on crank-pins are generally of two kinds—one running around the pin and the other running in direction of its length. The first kind is a dangerous one, and it must not be neglected. It is caused by the shaft coming inboard and throwing the thrust upon the crank, or by the shaft being out of line. In a case of this kind, drill a hole clear through the webs and pin—the larger the better—and put in a bolt, which must be a driving fit, rivetting over the ends. Those of the second kind—running lengthwise—are not so dangerous, and need only be chipped off and filed down.

BROKEN CRANK WEBS.

Cranks will sometimes show dangerous flaws and even break off. The remedy for this mishap would be to fit a stout strap or band around the crank, shrunk on and secured with tap bolts.

BROKEN SHAFTS.

Sometimes a shaft will break in such a way that it can be fixed up to run very easily by means of clamps; at other times the damage cannot be repaired at all with the material and tools to be had on ship-board.

BROKEN COUPLING BOLTS.

These are broken off, by reason of the shaft being out of line, when all go out of one coupling, and you have no extra ones on board, take one out of each of the other couplings and make up a set.

CIRCULATING PUMPS.

Should the circulating pump breakdown so that it cannot be repaired, turn the condenser into a jet condenser; means with which to do which are now furnished on all modern engines.

AIR-PUMP.

A broken air-pump is indeed a bad break on ship board. The best thing to do as a precaution is to carry duplicate parts of the piston

(bucket); some extra valves, so that new ones can at once be substituted for the broken parts.

BURSTED PIPES.

When a pipe bursts, place a piece of sheet iron smeared well over with white or red lead, over the crack and wrap it tightly all over with copper wire. In case of steam pipes use clamps—in halves—in addition to wire wrapping.

These are only a few of the many break-downs which may happen at sea under the best of management, and are only intended to give the learner a general idea of what is to be done in similar cases.

Engineers, like poets, are born not made, and a born engineer is always ready for any emergency—still prevention is better than cure in engineering as well as in physics; so prevent all the breakdowns you can.

REPAIRS TO BOILERS AT SEA.

Owing to imperfections in the iron, small cracks often appear in the crown-sheet, flues, and back connections. If in the crown-sheet, and not more than *two* inches in length, drill a hole, cutting the crack out, cut a taper thread, and insert a wrought-iron pipe plug (a number of which of different sizes should be carried for that purpose) smeared with white lead.

If in a flue, drill a number of small holes, and

put in small rivets, hammering the heads out so as to cover the crack.

If the crack is more than two inches long, or if it be a large hole which cannot be plugged, then you must put on a hard patch, and if the patch cannot be riveted on, then put it on with tap-bolts screwed into the sheet; care must be taken to admit water to the patch, else the fire will soon destroy it. Hard patches should be used on all parts of the boiler reached by the fire; and soft patches on the shell, and every other part not exposed to the fire.

The difference between a hard patch and a soft patch is, that a hard patch is *riveted* on or put on with tap bolts—bolts screwed into threaded holes both in the patches and sheet. A soft patch is put on with bolts and nuts. The best way to put on a soft patch is to first make a pattern of the patch needed out of sheet lead. Then take a piece of boiler plate and forge a patch as near like the lead pattern as possible, and lap over edge all around. Then drill as many ¾ holes as are necessary about 3 inches apart in the sheet first and corresponding ones in the patch afterwards: the holes should come true. Next mix some boiler-makers' cement which is composed of equal parts of white and red lead mixed with enough iron borings to make a stiff putty. Cover the inside of the patch with this putty and it is ready to go on. Now wrap some

5/8 bolts with lampwick smeared with plain white lead, pass the bolts through from the inside and set up the nuts tight, placing a washer between each nut and the patch. Should the patch be so situated that the bolts cannot be put through with the hand then they must be fished through. To fish a bolt you must take a long piece of wire, pass it through the bolt-hole and out through the hand-hole, then tie the bolt to the wire with a piece of strong string, giving about an inch slack, haul out the wire and the bolt will come out nicely in its place.

LEAKY TUBES, AND HOW TO PLUG THEM UP.

Let us suppose the tube is 3 inches in diameter, and 7 feet long. We take two white-pine plugs 3 inches in diameter, tapering off to 2 7-8 inches, and about 4 inches long. Next, bore a 3-4 inch hole through the centre of the plug. Take a 5-8 inch bar, 7 feet 4 inches long, with a thread cut two inches in length on each end. Run the bar through the tube, put in the pine plugs, and drive them in flush with the tube. Place a saucer-shaped washer filled with red lead, putty, or boiler-maker's cement, on each plug, and screw up tight with nuts.

The following cut will show at a glance how this is done:

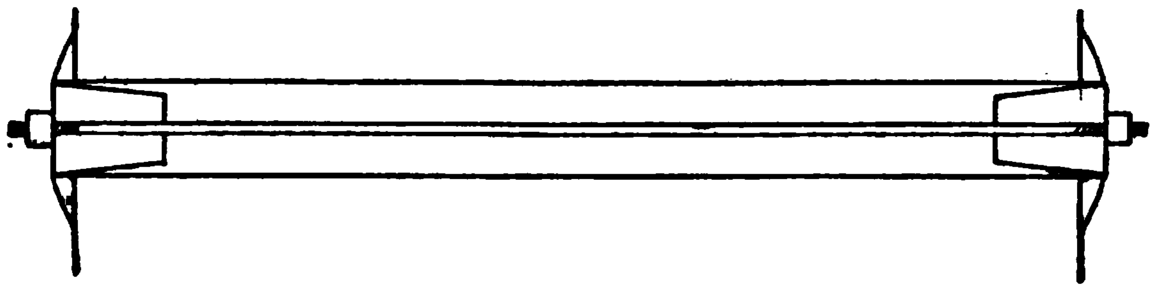

FIG. 72.

BEAM ENGINES (SIDEWEEL STEAMERS).

How to Set a Stevens Cut-off.

Let us suppose the eccentrics loose upon the shaft, and the toes loose upon the rock-shaft; now, also, let us suppose the engine should have 10 feet stroke, and we wish to cut off at half-stroke, that is, at 5 feet. We set the upper steam-valve first. To do this we measure off, on the slides from the top of the stroke, the distance of 2 1-2 feet, and then secure the cross-head just at that point. Next, block up the lifting rod with the *upper* valve its proper lift (1-4 its diameter), and secure it there by setting up on the binders. Next, secure the pin in the rock-shaft arm, about midway in the slot; then hook on the eccentric rod, and turn the eccentric forward its *full throw*, and then turn the toe on the rock-shaft up until it just touches the upper toe on the lifting-rod; secure the eccentric and toe on rock-shaft, and the *upper* valve is set to cut-off at half-stroke. To set the *lower* steam-valve, you merely turn the engine back until the eccentric gives its full throw backwards, and then

blocking up the lower steam-valve to its proper lift, you turn the other toe on the rock-shaft up until it just touches the toe on the lifting-rod, and secure it, and both your *steam*-valves are set. To set the exhaust-valves you place the engine on the centre, secure the pin in the rock-shaft arm, and also the toes, and then turn the eccentric around until it gives the proper lead, and then secure it.

The *steam*-valves on a beam engine require *no lead.* They should be set so as to just begin to lift as the crank passes the centre; but the *exhaust*-valves should have some lead. Poppet-valves set in this way as regards lead will give satisfaction.

The precaution must be taken in setting a Stevens cut-off not to raise the pin in the rock-shaft arm so that the eccentric will throw in a straight line, otherwise a break-down must follow. By lowering the toes on the rock-shaft arm it cuts off *shorter;* by raising them the steam follows further. After altering the position of the toes to give a different cut-off, the pin in the rock-shaft arm must be raised (or lowered as the case may be) to preserve the proper lift for the valve; and the eccentric moved forward to open the valve at the proper time.

A Stevens cut-off will not cut-off equally from both ends.

CHAPTER XXIII.

TAKING CARE OF AN ENGINE.

Finding the Cause of Thumping and Heating—The Constitutional Thumper.

If the bed plate of the engine be put down perfectly level, and securely bolted to a strong, solidly built hull, and the shaft be put down level, and exactly at right angles to the centre line of the cylinders of the engine, and the pillow blocks upon which the shaft rests upon solid foundations, and be well braced against the direction of the strain—in short, if the engine be properly put down, then it is the duty of the engineer who takes charge of, and runs it, to see that it is kept perfectly level, well bolted down and perfectly in line. If the bolts become so loose that the bed plate begins to move on the foundation, he should not screw them up until he examines carefully to see that the engine has not worked itself out of level or out of line.

The first thing likely to claim his attention is the proper adjustment of the brasses of the connecting-rod, and those on the journals of the shaft; these must fit their journals tight enough

to prevent thumping, and at the same time be loose enough to keep the journals from heating. This requires nice adjustment of the brasses. Just here generalities come to an end, and he must come down to the peculiar behavior of the one particular engine in his charge. Many times some of these brasses will heat, and at the same time thump badly; then he is apt to come to the conclusion that it is impossible for a journal to be too tight and at the same time too loose, but thinks that it is heating because it is too tight, and that the loose place or thumping is somewhere else. He slacks the box off just a very little to stop the heating, but when he starts up he finds that while the box is still heating a little, the thumping is a great deal worse than ever. Now he knows that the thumping is in that box, because he didn't alter anything else about the engine; but for the life of him he can't tell how it is, and every time he gets a chance to stop the engine he is tinkering with that box, tightening it a little this time, and slacking off a little the next, and concludes each time that he altered it too much, and so he worries along with it for four or five days, until finally it comes all right, quits heating and thumping. Then he is satisfied that he fixed it, but he cannot tell you how, and after awhile, when the same thing occurs again, he knows as little about it as before, and just tinkers away on it until it gets right again.

To find the cause is easy enough. If he will examine the box carefully he will find, strange as it may seem, that it really is both too tight and too loose; he will find that the key does not force the brasses up against the journal squarely, or rather the end of the connecting rod or the straps (as the case may be), does not fit squarely against the back of the brasses, and in that way the pressure against the journal is all on one side of the brasses, generally the bottom side, while there is space left between the back of the brass and the end of the rod at the top. When the engine is at work, this space is sufficient to allow it to thump, and in so doing the brasses are forced still tighter against the journal at the bottom, causing it to heat badly. So he finds that it was really too tight and too loose. He can make it adjust itself by tightening up on it, and letting it run until the tight side of the brasses wears down even with the slack side, but if he does this the brasses will be out of square on the side next to the journal, and will always be troublesome. The proper way to remedy it is to file off the brasses on the under side until they fit all over alike.

This condition is more often brought about by carelessness in putting in liners behind the brasses after they begin to wear. The average engineer is apt to make liners out of the first thing he can lay his hands on in the way of

sheet metal, generally a piece of hoop iron that is too narrow to cover more than one-half of the back of the brass, which, of course, throws it out of square; but he keys up on it and starts up his engine, and then wonders what makes the thing heat and thump so. He should be very careful to cut his liners out of sheet metal of even thickness, and cut them to fit and cover the whole back of the brass. The above applies equally well to the quarter boxing on the shaft.

There are engines that thump from the time they start up new until they are worn out; do what you will, you can't stop them from thumping, and you may send for the man who built the engine, and he can't stop it, and he cannot or will not tell you the cause of it. These are

Constitutional Thumpers,

and they are generally good drivers and last tolerably well if the man in charge of them don't tinker them to death trying to stop the thumping. Of course there is a cause for the thumping, but in order to correctly understand the cause it is necessary to take into consideration the direction of the greatest amount of strain brought to bear upon the frame-work of the engine while running under full steam. The steam at 150 pounds pressure discharged from the boiler rushes into the cylinder of the engine and against the piston, at the tremendous velocity

of 1450 feet per second, or at the rate of sixteen miles per minute, less the friction which would bring it down to say 1200 feet per second; the piston receives this tremendous shock just after it has reached the end of the cylinder, and is standing almost still; the crank pin is just crossing the centre, and is moving at right angles to the direction of the strain. So the whole strain of the shock is brought to bear upon the bed plate or framework of the engine. When the pin is on the top centre, the strain is upon the centre of the bed plate outward each way, and the next movement when it reaches the bottom centre the strain upon the bed plate is reversed. So that the strain brought to bear upon the bed plate or frame of a quick-working engine by these shocks, first in one direction and then in another, in quick succession, is tremendous. Now the constitutional thumper is an engine whose bed plate or frame is too weak, and each successive shock causes it to spring in the middle enough to throw the guides out of line, or gives the crosshead too much play in the guides, and *right there* you will find the thump of the constitutional thumper.

CHAPTER XXIV.

VALUABLE INFORMATION.

To find the capacity of a cylinder in gallons, multiply the area in inches by the length of stroke in inches and it will give the total number of cubic inches; divide this by 231 and you will have the capacity in gallons.

The U. S. standard gallon measures 231 cubic inches, and contains 8⅓ pounds of distilled water.

The mean pressure of the atmosphere is usually estimated at 14.7 pounds per square inch.

The average amount of coal used for steam boilers is 12 pounds per hour for each square foot of grate.

The average weight of anthracite coal is 53 pounds to one cubic foot of coal; bituminous, about 48 pounds to the cubic foot.

Locomotives average a consumption of 3000 gallons of water per 100 miles run.

To determine the circumference of a circle, multiply the diameter by 3.1416.

To find the pressure in pounds per square inch of a column of water, multiply the height of the column in feet by .434, approximately; every

foot elevation is equal to ½ pound pressure per square inch, allowing for ordinary friction.

The area of the steam piston, multiplied by the steam pressure, gives the total amount of pressure that can be exerted. The area of the water piston, multiplied by the pressure of water per square inch, gives the resistance. A margin must be made between the power and the resistance to move the pistons at the required speed, from 20 to 40 per cent., according to speed and other conditions.

To determine the diameter of a circle, multiply circumference by .31831.

Steam at atmospheric pressure flows into a vacuum at the rate of about 1550 feet per second, and into the atmosphere at the rate of 650 feet per second.

To determine the area of a circle, multiply the square of diameter by .7854.

FLOW OF STEAM THROUGH PIPES.

The quantity of steam flowing through a pipe in a given time under a given "head" or loss of pressure, increases more rapidly than the area of the pipe, or its ratio to the length, directly as the square root of the density, and of the loss of pressure, and inversely as the square root of the length.

The following table gives approximately the weight of steam per minute which will flow from

various initial pressures, with a loss of pressure due to friction of one pound per square inch, through different sizes of straight smooth pipes, each having a length of 240 times its own diameter (equal to 20 feet for each inch in diameter.) It has been calculated by the formula:

$$W = 303.25\, d^2 \sqrt{\frac{D\,(p_1 - p_2)}{L\left(1 + \frac{3.6}{d}\right)}}$$

In which d=diameter in inches, D=density or weight per cubic foot; p_1 the initial pressure, p_2 the pressure at end of pipe, and L=the length *expressed in diameters.* For sizes of pipe below 6 inch, the flow is calculated from the *actual* areas of "standard" pipe of such nominal diameters.

TABLE OF FLOW OF STEAM THROUGH PIPES.

Initial pressure by gauge. lbs. per sq. in.	Diameter of Pipe in Inches. Length of each = 240 diameters.													
	¾	1	1½	2	2½	3	4	5	6	8	10	12	15	18
	Weight of Steam per minute in pounds, with 1 pound loss of pressure.													
1 . .	1.16	2.07	5.7	10.27	15.45	25.38	46.85	77.3			341.1	502.4	804	1177
10 . .	1.44	2.57	7.1	12.72	19.15	31.45	58.05	95.8			422.7	622.5	996	1458
20 . .	1.70	3.02	8.3	14.94	22.49	36.94	68.20	112.6			496.5	731.3	1170	1713
30 . .	1.91	3.40	9.4	16.84	25.35	41.63	76.84	126.9			559.5	824.1	1318	1930
40 . .	2.10	3.74	10.3	18.51	27.87	45.77	84.49	139.5			615.3	906.0	1450	2122
50 . .	2.27	4.04	11.2	20.01	30.13	49.48	91.34	150.8			665.0	979.5	1567	2294
60 . .	2.43	4.32	11.9	21.38	32.19	52.87	97.60	161.1			710.6	1046.7	1675	2451
70 . .	2.57	4.58	12.6	22.65	34.10	56.00	103.37	170.7			752.7	1108.5	1774	2596
80 . .	2.71	4.82	13.3	23.82	35.87	58.91	108.74	179.5			791.7	1166.1	1866	2731
90 . .	2.83	5.04	13.9	24.92	37.52	61.62	113.74	187.8			828.1	1219.8	1951	2856
100 . .	2.95	5.25	14.5	25.96	39.07	64.18	118.47	195.6			862.6	1270.1	2032	2975
120 . .	3.16	5.63	15.5	27.85	41.93	68.87	127.12	209.9			925.6	1363.3	2181	3193
150 . .	3.45	6.14	17.0	30.37	45.72	75.09	138.61	228.8			1009.2	1486.5	2378	3481

For horse power, multiply the figures in the table by 2. For any other loss of pressure multiply by the square root of the given loss. For any other length of pipe, *divide 240 by the given length expressed in diameters, and multiply the figures in the table by the square root of this quotient*, which will give the flow for 1 lb. loss of pressure. Conversely, dividing the given length by 240 will give the loss of pressure for the flow given in the table.

The resistance to the steam entering the pipe, when it is not provided with special bell mouth opening, is equal to the friction of 60 diameters additional length, or 25 per cent. additional loss of pressure under the conditions of the table. A globe valve may be estimated as 60, and each elbow as 40 diameters additional length. Thus, a pipe 120 diameters in length, with one globe valve and three elbows, would have 1½ lbs. loss of pressure with the flow given in the table, or deliver $1 \div \sqrt{1\frac{1}{2}} = 81.7$ per cent. of the steam, with one pound loss of pressure. These equivalents, viz:—60 for opening, 60 for each globe valve, and 40 for each elbow, must be added in all cases in estimating the length of pipe; thus in the case just given, $120 + 60 + 60 + (3 \times 40) = 360 = 240 \times 1\frac{1}{2}$.

TABLE OF SPECIFIC GRAVITIES.

	Weight of a cubic inch in pounds.
Copper, cast	.3178
Iron, cast	.263
Iron, wrought	.276
Lead	.4103
Steel	.2827
Gun-metal	.3177

DIVISIONS OF DEGREES OF HEAT.

The thermometer is an instrument for measuring sensible heat. It consists of a glass tube of very fine bore, terminating in a bulb. This bulb is filled with mercury, and the top of the tube is hermetically sealed after all the air has been expelled. The instrument is then put into steam arising from boiling water, and when the barometer stands at 30 inches, a mark is placed on a scale affixed opposite the place the mercury stands at. It is again put in melting ice and the scale again marked. The space between these marks is divided into spaces called degrees. In this country and England it is divided into 180 equal parts, calling freezing point 32°, so that the boiling point is 212°, and zero or 0 is 32 below freezing point; and this scale is called Fahrenheit's. On the continent two other scales are in use: the Centigrade, in which the space is divided into 100 equal parts, hence the name; and Reaumur's, in which the space is divided

into 80. In both of these scales freezing point is 0 or zero, so that the boiling point of Centigrade is 100° and Reaumur 80°.

AVERAGE BREAKING AND CRUSHING STRAINS OF IRON AND STEEL.

Breaking strain of wrought iron = 23 tons . .	Per square inch of section.
Crushing strain of wrought iron = 17 tons . .	
Breaking strain of cast iron, about 7½ tons .	
Crushing strain of cast iron = 50 tons	
Breaking strain of steel bars about 50 tons . .	
Crushing strain of steel bars up to 116 tons .	

PROPORTIONS OF VARIOUS COMPOSITIONS IN COMMON USE. (IN 100 PARTS.)

Babbitt's Metal	Tin 89, Copper 3.7, Antimony 7.3.
Fine Yellow Brass . .	Copper 66, Zinc 34.
Gun Metal, Valves, &c.	Copper 90, Tin 10.
White Brass	Copper 10, Zinc 80, Tin 10.
German Silver	Copper 33.3, Zinc 33.4, Nickel 33.3.
Church Bells	Copper 80, Zinc 5.6, Tin 10.1, Lead 4.3.
Gongs	Copper 81.6, Tin 18.4.
Lathe Bushes	Copper 80, Tin 20.
Machinery Bearings .	Copper 87.5, Tin 12.5.
Muntz Metal	Copper 60, Zinc 40.
Sheathing Metal . . .	Copper 56, Zinc 44.

STEAM COAL.

Steam coal being, as everybody knows, unquestionably the most important and largest expense in the making of steam, is deserving a most careful investigation by engineers and owners, who, unlike chemists and college professors, consider the subject wholly in a practi-

cal way as relating to the coal bills of their establishments.

Useful knowledge of every-day economy of coal is seldom gained by "tests" conducted by experts, for several reasons so plain that they will not require explanation. 1st. The cost of the fuel used in "tests," whatever may be stated, is too high, average or "every-day" coal not being used. The experiments are made with picked men and picked fuel, for brief periods, with everything at its best, and the results obtained, if looked for in the ordinary run of business, will be disappointing in the results of the wholesale order. 2d. Men working as firemen in the hot fire rooms, cannot be expected, with the ordinary appliances, to watch where every lump of coal falls when feeding the furnaces, nor to clean the grates any oftener than they are compelled to do. 3d. Moreover, too many owners favor the low wages plan, and for an apparent saving of a few dollars per month, waste many times the amount in their furnace doors. 4th. Little or no encouragement is given for careful or economical firing, as a rule. The fireman who oftentimes wastes as much as his entire wages, secures the same pay as the man working alongside of him who saves it all. It may be remarked that this is "not business," but many are the concerns who run their vessels upon this system. Careful handling of coal in

firing pays better than any other thing about a steam vessel, and it is the wisest economy to secure good and careful men to do it.

As is well understood, the conditions or circumstances attending the combustion of coal for steam purposes, embrace a wide range. A very few steamers work under conditions that admit of a high attainment of economy. But by far the largest number have a fluctuating demand for steam, and in that respect are largely at a disadvantage. Many boilers are badly constructed, others suffer from an insufficiency of draft, and in many cases there seems to be no end of complications detrimental to best results.

These practical difficulties and uncertanties, which are well known to every experienced engineer, render any investigation worthy of the name slow and laborious. It has taken considerable time and research to arrive at the conclusion, though differing from the preponderance of hearsay or guess-work evidence, that now at least "*the highest priced coal is the cheapest for steam production*," Late improvements in the construction of grate-bars have undoubtedly added largely to the value of soft coals. The great difficulty in former times of ridding the furnaces of the incombustible part of these very valuable coals has now been removed by improvements.

AMERICAN COALS.

Coal. State.	Coal. Kind of Coal.	Per cent. of Ash.	Theoretical Value. In Heat Units.	Theoretical Value. In Pounds of Water evap.	Coal. State.	Coal. Kind of Coal.	Per cent. of Ash.	Theoretical Value. In Heat Units.	Theoretical Value. In Pounds of Water evap.
Penn.	Anthracite	3.49	14,199	14.70	Ill. .	Bureau Co.	5.20	13,025	13.48
" .	"	6.13	13,535	14.01	" .	Mercer Co.	5.60	13,123	13.58
" .	"	2.90	14,221	14.72	" . .	Montauk	5.50	12,659	13.10
"	. . . . Cannel	15.02	13,143	13.60	Ind. . . .	Block	2.50	13,588	14.38
"	Connellsville	6.50	13,368	13.84	"	. . . Caking	5.66	14,146	14 64
"	Semi-bi'nous	10 77	13,155	13.62	"	. . . Cannel	6.00	13,097	13.56
"	Stone's Gas	5.00	14,021	14.51	Md.	Cumberland	13.08	12,226	12.65
"	Youghiog'ny	5.60	14,265	14.76	Ark. . .	Lignite	5.00	9,215	9.54
"	. . . Brown	9.50	12,324	12.75	Col. . . .	"	9.25	13,562	14 04
Kent'ky.	Caking	2.75	14,391	14.89	" . . .	"	4.50	13,866	14.35
"	Cannel	2.00	15,198	16.76	Texas. .	"	4.50	12,962	13 41
" .	"	14.80	13,360	13.84	Wash.Ter.	"	3.40	11,551	11.96
"	Lignite	7.00	9,326	9.65	Penn.	Petroleum	. .	20,746	21.47

TEMPERATURE OF FIRE.

By reference to the table of fuels, it will be seen that the temperature of the fire is nearly the same for all kinds of combustibles, under similar conditions. If the temperature is known, the conditions of combustion may be inferred. The following table, from M. Pouillet, will enable the temperature to be judged by the appearance of the fire:

Appearance.	Temp. Fah.	Appearance.	Temp. Fah.
Red, just visible . . .	977°	Orange, deep.	2010°
" dull.	1290	" clear	2190
" Cherry dull . .	1470	White, heat	2370
" " full . . .	1650	" bright. . . .	2550
" " clear . .	1830	" dazzling. . .	2730

To determine temperature by fusion of metals, etc.

Substance.	Temp. Fah.	Metal.	Temp. Fah.	Metal.	Temp. Fah.
Tallow . . .	92°	Bismuth . .	518°	Silver, pure	1830°
Spermaceti.	120	Lead	630	Gold Coin	2156
Wax, white.	154	Zinc.	793	Iron Cast, medium.	2010
Sulphur. . .	239	Antimony. .	810	Steel.	2550
Tin	455	Brass	1650	Wrought Iron. . .	2910

FOAMING IN BOILERS.

The causes are dirty water; trying to evaporate more water than the size and construction of the boiler is intended for; taking the steam too low down; insufficient steam room; imperfect construction of boiler, and too small a steam pipe.

Take a kettle of dirty water and place it on a fire and allow it to boil and watch it foam, and it will be the same in a boiler.

Too little attention is paid to boilers with regard to their evaporating power. Where the boiler is large enough for the water to circulate, and there is surface enough to give off the steam, foaming never occurs. As the particles of steam have to escape to the surface of the water in the boiler, unless that is in proportion to the amount of steam to be generated, it will be delivered with such violence that the water will be mixed with it and cause what is called foaming.

A high pressure insures tranquility at the sur-

face, and the steam itself being more dense it comes away in a more compact form, and the ebullition at the surface is no greater than at a lower pressure. When a boiler foams we close the throttle to check the flow, and that keeps up the pressure and lessens the sudden delivery.

Too many tubes in a boiler obstruct the passage of the steam from the lower part of the boiler on its way to the surface; this is a fault in construction; but nearly all foaming arises from dirty water, or from trying to evaporate too much water without heating surface or steam room enough.

PROPERTIES OF SATURATED STEAM.

Pressure. By Steam Guage.	Pressure. Total.	Temperature in Fahrenheit Degrees.	Volume. Compared with Water.	Volume. Cubic Feet of Steam from 1 lb. of Water.	Latent Heat in Fahrenheit Degrees.	Total Heat required to generate 1 lb. of Steam from Water at 32 deg. under constant pressure.
						In Heat Units
0	15	212.0	1642	26.36	965.2	1146.1
5	20	228.0	1229	19.72	952.8	1150.9
10	25	240.1	996	15.99	945.3	1154.6
15	30	250.4	838	13.46	937.9	1157.8
20	35	259.3	726	11.65	931.6	1160.5
25	40	267.3	640	10.27	926.0	1162.9
30	45	274.4	572	9.18	920.9	1165.1
35	50	281.0	518	8.31	916.3	1167.1
40	55	287.1	474	7.61	912.0	1169.0
45	60	292.7	437	7.01	908.0	1170.7
50	65	298.0	405	6.49	904.2	1172.3
55	70	302.9	378	6.07	900.8	1173.8
60	75	307.5	353	5.68	897.5	1175.2
65	80	312.0	333	5.35	894.3	1176.5
70	85	316.1	314	5.05	891.4	1177.9
75	90	320.2	298	4.79	888.5	1179.1
80	95	324.1	283	4.55	885.8	1180.3
85	100	327.9	270	4.33	883.1	1181.4
90	105	331.3	257	4.14	880.7	1182.4
95	110	334.6	247	3.97	878.3	1183.5
100	115	338.0	237	3 80	875.9	1184.5
110	125	344.2	219	3.51	871.5	1186.4
120	135	350.1	203	3.27	867.4	1188.2
130	145	355.6	190	3.06	863.5	1189.9
140	155	361.0	179	2.87	859.7	1191.5
150	165	366.0	169	2.71	856.2	1192.9
160	175	370.8	159	2.56	852.9	1194.4
170	185	375.3	151	2.43	849.6	1195.8
180	195	379.7	144	2.31	846.5	1197.2

This table gives the value of all properties of saturated steam required in calculations connected with steam boilers.

COLLAPSE OF FURNACES.

"In marine boilers one of the most frequent, most annoying, and at the same time most expensive accidents (?) is the collapse of the furnaces. Those interested may possibly be pleased to learn that extensive experiments have recently been made by M. Hirsch on the causes which lead to the burning out of furnace plates, and as a result he confirms the fact, long known to experts, that under some conditions oil in the interior of the boiler is highly dangerous. The interior of a boiler to be examined was painted with oil before being filled with water and firing in the usual way, and it was found that some oils so diminished the efficiency of contact between the water and the plates that in one case the furnace plates rose to a temperature exceeding 680°, or the melting point of zinc, when only evaporating 35 lbs. of water per square foot of grate surface."

The combustion chamber should be larger where bituminous coal is used.

In piping boilers avoid too rigid connections, and thus prevent leakage and accidents.

The transverse strain in a cylinder boiler is double the longitudinal strain, and in a tubular boiler a much larger ratio exists, because of the support given the heads by the tubes.

The smallest point in a suction pipe is practically the measure of ability of the whole; hence

where a foot valve is used, care should be taken that it has as large an opening as the pipe itself.

In careful engine tests a counter can be set up and connected with the reducing motion and the speed of the engine thus noted. The reading can be divided by the exact length of test and an accurate average thus obtained.

INCRUSTATION OF STEAM BOILERS.

One of the greatest difficulties to be contended against in steam engineering is the incrustation of the boiler walls, arising from impure water. This crust is a poor conductor of heat, and causes increased fuel consumption, as well as the oxidizing or "burning" of the plates, owing to their increased temperature. A plate of iron 37½ inches thick conducts heat as well as a "crust" of one inch. A boiler bearing scale only 1/16 inch thick requires 15 per cent. more fuel, with ¼ inch 60 per cent. more, ½ inch 150 per cent. more. If the plates be clean, 90 pounds of steam require a plate temperature of only 325° F., that is, about 5° above the steam temperature. But if there be a ½ inch scale or crust, the plate must be heated to about 700°, or nearly "low red" heat. Now, above 600° iron soon gets granular and brittle, hence, such a scale is dangerous in its results. Crust also retards the circulation of the water. Two very common ingredients in boiler scale are carbonate

of lime and sulphate of lime, or gypsum. The moderate use of soda ash (say one part in 5,000 of water) holds this deposit in check, by producing from the principal ingredient a *neutral* carbonate of lime, which will not adhere to the plates, when thus rapidly formed. Soda ash, if used in excess, boils up and passes into the cylinders and pumps, clogging up valves and pistons by combining with the lubricants. If the gauge-glasses become muddy, too much soda water is used. It is much better to supply the boilers with pure water that can deposit no scale, this being done by means of filters and heaters, or by surface-condensers, and being especially advisable with sectional and water tube boilers.

ANALYSIS OF BOILER INCRUSTATION.

BY DR. WALLACE.

Carbonate of lime	64.98
Sulphate of lime	9.33
Magnesia	6.93
Combined water	3.15
Chloride of sodium	.23
Oxide of iron	1.36
Phosphate of lime of alumina	3.72
Silica	6.60
Organic matter	1.60
Moisture at 212° F.,	2.10
	100.

HOW TO PREVENT ACCIDENTS TO BOILERS.

1st. Carry regular steam pressure.

2d. Start the engine slowly, so as not to

make a violent change in the condition of the water and steam, and when consistent stop the engine gradually.

3d. Carry sufficient water in the boiler.

4th. Do not exceed the pressure in lbs. per square inch allowed to be carried.

5th. See that every appliance of the boiler—feed pipes and safety-valve, fusible plugs, etc.—are in complete working order.

WEIGHT OF DIFFERENT SUBSTANCES.

The following table shows the weight in pounds of a cubic foot of various common substances. From this table we can calculate the weight of any mass of any substance in the table, by first ascertaining by measurement and calculation how many cubic feet there are in the mass, and multiplying the result by the weight of a cubic foot of the substance:

SUBSTANCE.	Pounds per cubic foot.
METALS.	
Cast iron	450.5
Wrought iron bar	486.6
Soft steel	489.8
Common brass	537.7
Wrought copper	555.
Cast lead	709.
Cast zinc	428.8
Tin	456.
Mercury, at 60°	848.75
Gold	1013.
Silver	551.

SUBSTANCE.	Pounds per cubic foot.
Antimony	422.
Aluminium	160.
Platinum	1296.

WOODS.

Oak	55.
Ash	53.
Cork	15.
Cedar	35.
Chestnut	38.
Hickory	45.
Lignum-vitæ	83.
Mahogany	50.
Pine, yellow	34.
Spruce	31.
Walnut	31.
Willow	30.

MISCELLANEOUS.

Air	00.075
Brick	102.
Coal, soft	80.
Coke	62.
Pressed cotton	22.
Gutta-percha	61.
Ice, at 32°	57.5
Moist sand	128.
Common earth	137.
Gravel	126.
Glass, window	165.
Granite	165.
Grindstone	133.
Marble	169.
Millstone	155.
Mud	101.

SUBSTANCE.	Pounds per cubic foot.
LIQUIDS.	
Alcohol	50.
Milk	64.
Petroleum oil	55.
Water, rain	62.5
Water, Dead Sea	77.5

STRENGTH OF SUBSTANCES.

The following table shows the breaking strength of, or power in pounds required to pull apart a bar one inch square of various common substances. The strength of any piece of substance shown in the table can be found by measuring or calculating the number of square inches area of the piece and multiplying the result by the breaking strength of one square inch, as shown in the table:

SUBSTANCE.	POUNDS.
Wrought copper	34,000
Copper wire	61,200
Cast iron—American	31,800
Cast iron—English	19,500
Iron wire	103,000
English bar	56,000
Boiler plate	50,000
Cast lead	1,800
Platinum wire	53,000
Cast silver	40,000
Bar cast steel	88,000
Shear steel	124,000
Cast tin	5,000
Zinc, cast	3,500

SUBSTANCE.	POUNDS.
Hydraulic cement	234
Glass	2,350
Gutta-percha	3,500
Ivory	16,000
Leather belts	330
White marble	9,000
Mortar, good	60
Manila rope	9,000
Hemp rope	15,000
Wire rope	37,000
Stone	350
Gold	40,000
Brass	42,000
Bronze, best	56,800
Oak	10,000
Boxwood	20,000
Beech	19
Maple	44
Yellow pine	14
Cedar	13
White pine	11
Hemlock	8
Cork	5

HARDNESS OF SUBSTANCES.

It is, of course, impossible to speak of hardness in pounds, or square feet, or anything of that kind. We can only compare the hardness of one substance with the hardness of another substance. Thus, we can say that chalk is half as hard as gold. It is best to take some one substance as a standard and compare other substances with it. Taking Ormuz diamond as the standard, we find cork to be one-twentieth as

hard. For convenience it is best to give the standard a value of one hundred, and express the hardness of other substances in hundredths of the standard. The following table shows the relative hardness of various substances, Ormuz diamond being taken as a standard:

SUBSTANCE.	HARDNESS.
Ormuz diamond	100
Agate	71
Quartz	70
Ruby	64
Granite	22
Chalk	15
Hard steel	65
Unhardened steel	40
Cast iron	38
Wrought iron	37
Platinum	47
Common brass	32
Hammered gold	30
Hammered copper	34
Hammered silver	32
Zinc	26
Aluminium	24
Tin	24
Babbitt metal	20
Cast lead	15
Ebony	24
Boxwood	22
Oak, hard	22

SHRINKAGE OF CASTINGS.

In steam cylinders, $\frac{1}{16}$ inch in a foot.
Pipes, $\frac{1}{8}$ inch in a foot.

Girders, beams, etc., 1/8 inch in 15 inches.

Engine beams, connecting rods, etc., 1/8 inch in 16 inches.

Large cylinders, say 70 inches diameter, 10 feet stroke, the contraction of diameter, 3/8 inch at top.

Ditto, 1/2 inch at bottom.

Ditto in length, 1/8 inch in 16 inches.

Thin brass, 1/8 inch in 8 inches.

Thick brass, 1/8 inch in 10 inches.

Zinc, 5/16 inch in a foot.

Lead, 5/16 inch in a foot.

Copper, 3/16 inch in a foot.

Bismuth, 5/32 inch in a foot.

Tin, 1/4 inch in a foot.

CHAPTER XXV.

STEAM YACHTS AND LAUNCHES.

HITHERTO the design and construction of a steam yacht has been carried on in a hap-hazard sort of way, that is only too apparent from a casual inspection of the vessel when completed. One firm is called upon to whittle out a model of a certain length, another is asked to build the engine, while a third supplies a boiler of certain dimensions. The interior fittings are supplied by individuals, perhaps totally unacquainted with marine work. Each individual concerned may be competent to carry out his own particular part of the work, but he pays but little attention to any other portion, and the result, while perhaps correct in detail, is decidedly lacking in harmony.

The first requisite is speed, and accommodation comes next in the scale. The former can only be obtained by an easy model and great power, while the latter demands close attention to the utilization of space. In order to produce an average of these essentials, the engine and boiler must be as small as possible, while at the same time capable of exerting great power, and so designed as to be free from vexatious

break-downs under continuous work. The consumption of fuel should be reduced to a minimum in order to economize space, and to avoid the inconvenience of frequent coaling.

In addition to these requisites a certain amount of *beauty* is necessary in order to harmonize with the purpose for which the vessel is to be used. The most important point in naval design is *harmony of details*, and this can only be attained by the influence of *one competent mind.* Errors in steam launches resulting from incorrect design are of comparatively less importance, but in steam yachts, where the amount of money involved is much greater, it is important that the design should be correct.

DIMENSIONS OF SMALL STEAM YACHTS AND LAUNCHES.*

NUMBER.	1	2	3	4	5	6	7
Length of Boat, over all . . .	21	25	30	33	40	45	50
Breadth outside.	5	6	7	7½	8	8.6	10
Depth amidship.	2.4	2.9	3.0	3.0	3.0	3.3	4.6
No. of persons will carry with comfort and safety	9	15	25	30	35	40	60
Approximate speed, miles per hour . . .	7	8	9	10	10½	10	11½
Approximate consumption of soft coal per 10 hours, lbs. . .	150	250	350	400	600	750	1,000
Actual Indicated Horse Power, with 100 lbs. steam pressure.	3½	8	10	12½	18	24	32½
Engine, Diameter Cylinder.	3	4½	5	6	6	7	8
Engine, Length of Stroke .	4	5	7	8	8	8	10
No. of Revolutions per min.	300	300	275	275	250	250	200
Vertical Boiler, Diameter . . .	22	26	30	36	40	42	48
Vertical Boiler, Height	36	48	48	50	53	58	72
Propeller Wheel, Diameter . . .	20	24	30	36	36	38	42
Wheel Shaft, Diameter	1¼	1½	1¾	$1\frac{15}{16}$	$1\frac{15}{16}$	$2\frac{3}{16}$	$2\frac{7}{16}$

* Chas. P. Willard & Co., Chicago, Ill.

DIMENSIONS FOR A SMALL STERN WHEEL STEAMER.*

Length of hull, feet	33
Length of boat, feet over all	40
Beam, outside, feet	8
Depth, feet	3
Draft, inches	16
Number of persons will carry with comfort	30
Approximate speed, miles per hour	10
Approximate consumption of soft coal per 10 hours, pounds	400
Actual Horse Power with 70 pounds of steam	10
Engines—two in number—diameter of cylinder, inches.	5
Engines—two in number—stroke, inches	20
Vertical boilers, diameter, inches	40
Vertical boiler, height, inches	53
Diameter of stern-wheel, feet	7

* Willard.

CHAPTER XXVI.

MODERN AMERICAN MARINE ENGINES AND BOILERS, DESIGNED BY THE BUREAU OF STEAM ENGINEERING U. S. NAVY DEPARTMENT—CHIEF ENGINEER GEORGE W. MELVILLE, U. S. N., CHIEF OF BUREAU.

Description of Machinery Designed by the Bureau for New Vessels.

THE following descriptions have been prepared in the belief that they will be of general interest.

Maine.

The propelling engines are of the twin-screw, vertical, triple-expansion type in separate watertight compartments. The cylinders are 35½, 57, and 88 inches in diameter, by 36 inches stroke, and are to work at 132 revolutions per minute. The collective I. H. P. of the main and the air and circulating pump engines at full power is to be 9,000.

The principle of interchangeability of parts, which in many lines of manufacture has revolutionized modern practice, is in these engines carried out to the fullest extent.

All the cylinders have piston-valves of the same size—22 inches diameter—there being one for the H. P., two for the I. P., and three for the L. P. cylinder. The valves for each cylinder are worked from a Stephenson double-bar link. The cylinders have working linings of hard cast-iron and are steam-jacketed.

The H. P. and L. P. cylinders rest upon hollow cast steel columns, and the I. P. cylinder upon straight columns in front and inverted **Y** columns at the back. The columns are firmly bolted to the cylinders, to the cast-steel bed plate, and to each other. The crank shafts are of forged steel, 13 inches external diameter at the journals, with axial holes 4 inches in diameter through shafts and crank pins, which latter are 14 inches diameter. Each shaft is in three sections which are interchangeable. The thrust shafts are 12¾ inches diameter, with 6-inch axial holes, and the propeller shafts 13¼ inches with 6½ and 6-inch axial holes.

The condensers are cylindrical and of composition, $\frac{7}{16}$ inch thick. They are 6 feet 5½ inches in internal diameter, and the tubes 8 feet 4 inches long, with a condensing surface in each condenser of 7,010 square feet. Each circulating pump is centrifugal, has a capacity of 8,000 gallons per minute, and is fitted with a bilge connection for use as a wrecking-pump. For each condenser there are two double-acting hori-

zontal air-pumps, 17½ inches diameter and 18 inches stroke, driven by a vertical compound engine. The air-pump connecting rods take hold of the same crank-pins as the engine connecting rods. A valve is fitted in each L. P. exhaust pipe, so as to shut off the commnnication with the condenser and permit it to be used for auxiliary purposes when the main engines are stopped.

There are eight single-ended boilers of the ordinary return tubular type, 14 feet 8 inches diameter and 10 feet long, designed for a working pressure of 135 pounds. The shell plates are of mild steel, 1 1/16 inches thick. Each boiler has three corrugated steel furnaces 42 inches diameter, and 519 steel tubes 2¼ inches in external diameter and 6 feet 7 inches long. There are 118 stay tubes and 401 plain tubes in each boiler. The ordinary tubes are No. 12 and the stay tubes No. 6 B. W. G. thick. The total grate-surface is 553 square feet, and the total heating surface about 18,800 square feet. The boilers are in two separate water-tight compartments with fore and aft fire rooms. There are two fixed smoke-pipes about 60 feet in height above lower grates. The forced draft is on the closed ash-pit system, the air being led into the ash-pits by ducts under the fire-room floors. The blowers are driven by inclosed three-cylinder engines, and are arranged to draw the air

from the engine and fire-rooms so as to give thorough ventilation.

The main and auxiliary feed systems are duplicates as to pumps, piping, and valves, except that the main feed-pump can draw water only from the feed-tanks, while the auxiliary feed-pumps can also draw from the sea. Provision is made for heating the feed-water.

Evaporators are fitted for furnishing the steam for the distillers and for replenishing the fresh water lost in ordinary running. Connections are fitted from the evaporator to the auxiliary exhaust mains, so that the steam can go direct to the condensers. The distilling plant has a capacity of 5,000 gallons of potable water per day at a temperature of not over 90°, but when the evaporators are used for supplying losses the capacity is greatly increased. Spare coils are supplied for the evaporators, so that clean ones can be inserted when necessary to remove the scale from those which have been in use.

The auxiliary machinery includes reversing and turning engines, fire, bilge, drainage and flushing pumps, ash-hoists, workshop tools, etc.

A special feature of the propelling engines of this ship is the arrangement for disconnecting the low-pressure cylinder when cruising at low speeds. These are placed forward and provision is made for readily disconnecting their crankshafts. This reduces each engine to a two-

cylinder compound which can work up nearly to its full power, thus giving an increased economy over the triple-expansion engine at very low powers. A special exhaust leads from the intermediate cylinder to the condenser for use in such cases.

The propellers are to be of manganese or aluminum bronze, three bladed and about 15 feet diameter.

Besides the steam machinery there is also a complete hydraulic plant for operating the turrets, loading and turning the guns, etc. The working pressure is 600 pounds per square inch.

Monadnock.

The machinery now building at the Mare Island navy-yard for this vessel consists of twin-screw, horizontal, direct-acting, triple-expansion engines designed to develop with the air and circulating pumps about 3,000 I. H. P. when making 150 revolutions per minute with steam of 160 pounds pressure.

The engines are entirely independent and in separate water-tight compartments. The H. P. cylinder is aft in the forward compartment and forward in the after compartment. The forward engine drives the port propeller.

The cylinders are 19¾, 30¾, and 52¾ inches diameter by 30 inches stroke. The main valves are of the piston type worked by double bar

links, there being one of 11 inches diameter for the H. P. cylinder, and one of 15 inches diameter for the I. P. cylinder, and two of 15 inches diameter for the L. P. cylinder, those for the I. P. and L. P. cylinders being interchangeable.

The cylinders are tied to the crank-shaft pillow-blocks by forged steel tie-rods and the cast-iron cross-head guides. The latter are of the bar type, hollow, and provided with water circulation. The craft-shafts are of forged steel in three interchangeable sections. They are 9 inches diameter at the journals and 9½ inches for the crank-pins, with 3-inch axial holes. The cranks are at angles of 120°. The line shafts are of forged steel 8½ inches diameter, with 4½-inch axial holes. The thrust shafts are 9 inches diameter, with 4½-inch axial holes. The old propeller shafts of wrought iron are in the vessel and will be used.

A combined steam and hydraulic reversing gear, steam starting valves, and steam turning engines are fitted.

The condensers are cylindrical, of sheet-brass and composition, with 2,485 square feet of cooling surface in each. A valve is fitted in each low-pressure exhaust-pipe, so as to shut off the cylinder when the main engines are stopped and the condenser used for auxiliary purposes. Centrifugal circulating-pumps, each with a

capacity of 3,000 gallons per minute when used for wrecking purposes, are supplied, one for each condenser. Each condenser has two vertical single-acting air-pumps with cylinders 13½ inches diameter by 12 inches stroke, driven by a compound engine, the steam and pump pistons being on the same piston-rods. A connecting rod drives a shaft with fly-wheel in the center. This makes a very neat and compact arrangement.

There will be four boilers of the ordinary cylindrical type, 12 feet 2 inches diameter and 10 feet 1 inch long, each having two 46-inch corrugated furnaces. The total grate surface is 200 square feet and the total heating surface 6,242 square feet. The boiler shells are of mild steel 1 5/64 inches thick. The size, material, and spacing of tubes is the same as in the *Maine's* boilers. The ends of the stay tubes in the combustion chambers are protected by cast-iron ferrules, and those of the plain tubes by cast-iron rings or washers. The closed ash-pit system of forced draft will be used, the blowers being driven by inclosed multiple cylinder engines. The boilers are placed in two separate water-tight compartments, with fore and aft firerooms.

The feed systems are like those of the *Maine*, but instead of hydrokineters for circulating the water while raising steam, connections are made

from the bottom blow-pipes to the auxiliary feed-pump, so that this pump can be used for the purpose. Provision is made for heating the feed-water.

Evaporators with a capacity of 3,000 gallons of potable water per day are to be fitted with connections to the auxiliary exhaust mains for supplying the loss of fresh water.

There will be the usual auxiliary machinery and tools in workshop.

Armored Coast-Defense Vessel.

Twin-screw vertical triple expansion engines of 5,400 I. H. P. have been designed for this vessel.

The revolutions will be 150 per minute and the steam pressure 160 pounds. The cylinders are 27, 41, and 64 inches diameter and 30 inches stroke. In their general features these engines resemble those of the *Maine*, except the provision for disconnecting the L. P. cylinder. The H. P. cylinder is forward in each engine. The H. P. has one piston-valve 14 inches in diameter, the I. P. two of 14 inches, and the L. P. two of 20 inches, all worked by Stephenson double-bar links. The cylinders are supported by cast-steel inverted Y frames secured to cast-steel bed-plates.

The crank-shaft is of forged steel, in three interchangeable sections, with 4-inch axial holes

through shafts and crank-pins. The journals and crank-pins are 11 inches diameter. The line, thrust, and propeller shafts will be 10 inches diameter, with a 4-inch axial hole.

The screw propellers will be of manganese or aluminum bronze, three-bladed and about 10 feet 6 inches diameter. The starboard one will be right, and the port left handed.

The condensers will be cylindrical, of composition, with about 3,850 square feet of cooling surface in each. The circulating-pumps will be centrifugal, with a capacity of 5,000 gallons per minute each and connections for working as wrecking-pumps. Each condenser will have two vertical single-acting air-pumps, 14⅜ inches diameter by 15 inches stroke, driven by a compound engine with a fly-wheel at each end of shaft. A valve is provided in the exhaust-pipe from each L. P. cylinder, so as to shut off the connection to the condenser and permit it to be used as an auxiliary condenser when the main engines are stopped.

The engines are fitted with starting-valves, a steam-actuated throttle, and a combined steam and hydraulic reversing-gear, so that they can be handled with ease.

The usual auxiliary engines are fitted.

In this ship it was desirable that the weight of machinery should be reduced to the lowest limit. The engines having been made as light

as possible, the only room for reduction was in the boilers. The Bureau therefore decided, after mature consideration, to advise that about three-fourths of the boiler-power should be put in the form of coil or tubulous boilers. The advantage of these boilers in point of lightness and rapidity of raising steam has long been appreciated, but a distrust of their durability and proper functioning when arranged in group has existed. While there are sufficient grounds for this feeling in the case of many boilers of this type, the Bureau is satisfied that there are some which will work well, and, as stated elsewhere, this will soon be determined by the Board on Coil Boilers.

The service which this vessel is to perform, coast defense, makes boilers of this type especially useful. Ordinarily she will be at anchor or under easy steam, the two cylindrical boilers which are to be fitted giving steam for ten knots speed. The emergency power in the coil boilers will enable steam to be raised in less than half an hour, in sufficient quantity to give seventeen knots.

If, as the Bureau anticipates, these boilers give entire satisfaction in practice, we shall be able to reduce the boiler weights for future ships almost one-half.

The two cylindrical boilers which are to be fitted are to work at 160 pounds, and are 11 feet

2 inches diameter by about 10 feet 7 inches long, the shell-plates being of mild steel $\frac{63}{64}$ inch thick. There will be two corrugated furnaces in each boiler, 42 inches diameter. Each boiler will have about 44 square feet of grate surface and about 1,420 square feet of heating surface. The material, size and arrangement of tubes is the same as already described for the *Maine* and the *Monadnock*.

As in the other vessels, evaporators will be fitted for supplying the distillers and replacing losses of fresh water. The evaporators are to have a capacity sufficient to supply in twenty-four hours the water necessary to fill the coil boilers.

Provision is made for heating the feed-water. The feed systems are the same as in the other vessels.

Cruisers 7 and 8, of 3,000 Tons Displacement.

These vessels are designed for a very high speed, and in consequence very powerful engines are needed to secure it. They are to be twin-screw, vertical, triple expansion engines of 10,000 I. H. P. at full power, when making 164 revolutions with 160 pounds pressure. The cylinders are 36, 53, and two of 57 inches diameter, by 33 inches stroke. Two low-pressure cylinders are fitted, because of the limited space athwartship, which would not have permitted a

good arrangement with a single large cylinder. Each engine is in a separate water-tight compartment. The piston-valves are all 20 inches diameter, there being one for each H. P. cylinder, two for each I. P. cylinder, and two for each L. P. cylinder. They are all worked from Stephenson double-bar links. Provision is made in these engines, as in all the Bureau's recent designs, for adjusting the point of cut-off for each cylinder independently of the others, by making the attachment of the suspension-rod of the link to the arm on reversing-shaft adjustable. The crank-shafts are in three sections, the two forward ones being interchangeable and the after ones reversible. The two L. P. cranks are placed opposite each other, as are the H. P. and I. P. cranks, the plane of these two being at right angles to that of the two L. P. cranks. The journals are 13½ inches diameter, and the crank-pins 14½ inches diameter, all with 6-inch axial holes. The thrust shafts will be 13 inches diameter with 6½-inch axial holes, and the propeller-shafts 13¼ inches diameter with 6½ and 6-inch axial holes. The propellers will be three-bladed, of manganese bronze, or equivalent metal, and about 14 feet 6 inches diameter.

The condensers will be cylindrical, of composition, 5 feet 8 inches diameter, and the tubes 11 feet 6 inches long, each having a cooling surface of about 6,990 square feet. A valve is

fitted in the exhaust-pipes from L. P. cylinders to shut them off when the condenser is used for auxiliary purposes. Each centrifugal circulating pump will have a capacity of 9,000 gallons per minute when pumping from the bilge. There will be two vertical single-acting air-pumps for each condenser, 18½ inches diameter and 16½ inches stroke, worked by a compound engine.

There will be steam starting-valves, steam-actuated throttle-valves, steam and hydraulic reversing-engine, turning-engine, workshop machinery, and the usual auxiliaries.

There will be four double-end and two single-end boilers of the usual return tubular type, all built of mild steel. Two of the double-end boilers will be 13 feet 4 inches diameter, and two 14 feet 6½ inches diameter, all 20 feet 3½ inches long. The single-end boilers will be 11 feet 6 inches diameter by 9 feet 10¼ inches long. The shell-plates will be $1\frac{5}{32}$ inches, $1\frac{15}{64}$ inches, and $\frac{57}{64}$ of an inch, respectively. The double-end boilers will each have six corrugated furnaces 44 inches diameter for the large, and 40 inches diameter for the small ones. The single-end boilers will each have two corrugated furnaces 42 inches diameter. The total grate surface is 607 square feet, and the total heating surface 20,167 square feet.

The size, material, and spacing of tubes are the same as for the *Maine's* boilers. The feed systems are also like those of that ship.

The same arrangement of the evaporators for the various purposes exists as in the other ships, the capacity being 5,000 gallons of potable water per day.

Cruisers 9, 10 and 11, of 2,000 Tons Displacement.

The machinery for these vessels is, in its general features, a reduced copy of that for the 3,000-ton vessels, but each main engine has only one L. P. cylinder. The dimensions are as follows: Cylinders, 26½, 39, and 63 inches diameter by 26 inches stroke. I. H. P. of main engines and air and circulating pumps at full power, 5,400, with 185 revolutions and 160 pounds pressure.

The piston-valves are one for the H. P. cylinder 14 inches diameter, one for I. P. cylinder 14 inches diameter, and two for L. P. cylinder 20 inches diameter.

The journals and pins of crank-shaft are 10 inches diameter with a 5-inch axial hole. The three sections are interchangeable, and the cranks are set 120° apart. The line and thrust shafts are 9¾ inches diameter with a 5-inch axial hole, and the propeller-shafts are 10½ inches with a 6-inch axial hole.

The propellers will be four-bladed, about 11 feet diameter, and of manganese bronze or equivalent metal.

The condensers have each a cooling surface of about 3,957 square feet.

The centrifugal circulating-pumps have each a capacity, when pumping from the bilge, of 5,500 gallons per minute.

The air-pumps are of the same type as for the 3,000-ton cruisers, and have a diameter of 14⅜ inches and a stroke of 15 inches.

There are to be three double-end and two single-end boilers of the ordinary cylindrical type, all 11 feet 8 inches diameter. The forward double-end boilers are 18 feet 8½ inches long, the after double-end boiler is 18 feet 1 inch long, and the single-end boilers about 9 feet long. The shell-plates are $1\frac{1}{64}$ inches thick. The total grate surface is about 336 square feet, and the total heating surface 10,966 square feet.

The forced draft is on the closed ash-pit system.

Armored Cruising Monitor.

The engines for this vessel are of the same general type as those for the *coast defense vessel.* They are twin-screw vertical triple-expansion engines in separate water-tight compartments. The cylinders are 31¼, 46, and 70 inches diameter, by 36 inches stroke. The I. H. P. of the main engines and air and circulating pumps is to be 7,500 at 150 revolutions per minute and 160 pounds working pressure.

The piston-valves worked from Stephenson double bar links will have the following diameters: H. P. one of 15 inches; I. P. two of 15 inches; L. P. three of 20 inches. The valves and valve gear of the H. P. and I. P. cylinders are to be interchangeable. The crank-shaft journals will be $12\frac{1}{2}$ inches diameter, and the crank-pins 13 inches diameter with 6-inch axial holes. As in the *coast defense vessel*, the sections of the shaft are interchangeable and the cranks at 120°. The line, thrust, and propeller shafts will be about $12\frac{1}{4}$ inches diameter with 6-inch axial holes.

The condensers will have about 5,237 square feet of cooling surface in each. The centrifugal circulating pumps are to have a capacity as wrecking pumps of about 7,000 gallons each. The two air-pumps for each condenser will be 16 inches diameter by 16 inches stroke.

There will be eight single-ended cylindrical boilers, 12 feet 6 inches diameter and 10 feet 11 inches long, each having three-corrugated furnaces 36 inches diameter. The total grate surface is about 444 square feet, and the total heating surface 15,050 square feet.

The boiler shells are $1\frac{1}{32}$ inch thick.

The forced draft is on the closed ash-pit system.

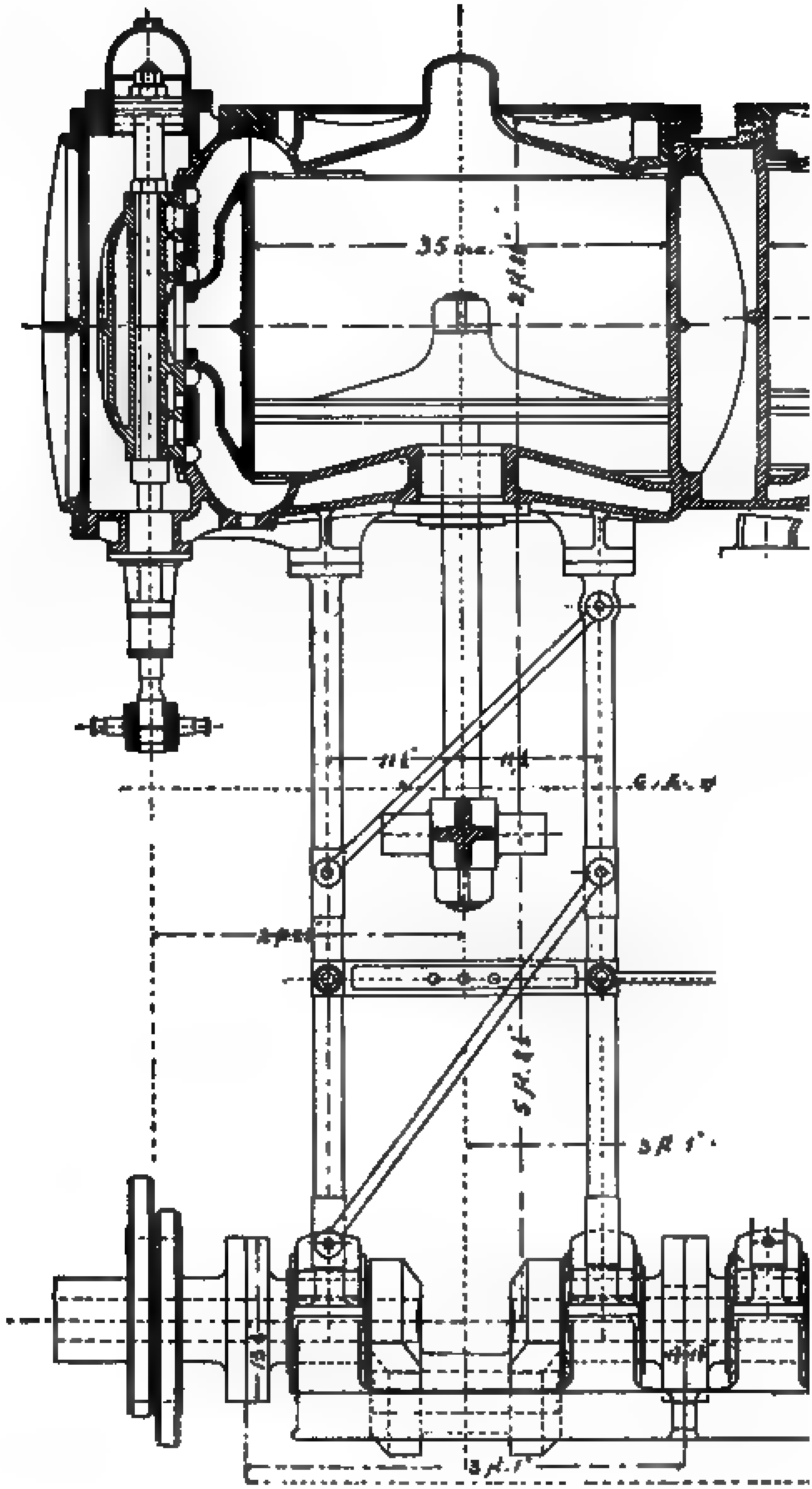

FIG. 73.—TRIPLE EXPANSION ENGINES OF NEW U. S. CRU
GEO. W. MELVILLE, U. S. NAVY, CHIEF OF BUREA

ISERS NOS. 12 AND 13. DESIGNED BY ENGINEER-IN-CHIEF OF STEAM ENGINEERING, U. S. NAVY DEPARTMENT.

TO FACE PAGE 368.

Cruisers 12 and 13 of 1,000 Tons, and Naval Academy Practice Vessel.*

The machinery of these vessels is similar in all details. They will be propelled by twin-screw, vertical, triple-expansion engines, placed in a common water-tight compartment. The two sets of engines are independent, except that there is only one condenser for both. An auxiliary condenser with a capacity equal to all the auxiliary machinery, except the main air and circulating pumps, is to be supplied. This will have an independent, combined air and circulating pump. The main condenser will have air-pumps and a circulating pump similiar in general features to those for the *Monadnock.*

The valves for the H. P. and I. P. cylinders will be plug pistons grooved, one for the H. P. and two of the same size for the I. P., while the L. P. cylinders will have double ported slide valves, all worked by Stephenson double-bar links.

The boilers will be of the low cylindrical type, all in one water-tight compartment.

Evaporators, distillers, auxiliary machinery, etc., are provided as in the larger ships. Provision is made for heating the feed water.

The following are the dimensions of the machinery: For *cruisers* 12 and 13, cylinders,

*See drawings. Figs. 73, 74, 75, 76.

15¾, 22½, and 35 inches diameter by 24 inches stroke. I. H. P. 1,600 at 200 revolutions per minute and 160 pounds working pressure. Piston-valves for H. P. and I. P. cylinders, 7½ inches diameter. Diameter of crank-shaft journals and crank-pins, 7 inches, with 3½-inch axial holes. Cranks at 120°, and sections of shafts interchangeable. The line, thrust, and propeller shafts will be about 7 inches diameter, with 3½-inch axial holes.

Cooling surface in main condenser about 2,246 square feet. Diameter of air-pumps 13½ inches, stroke 12 inches. Capacity of circulating pump 3,000 gallons per minute.

There will be two boilers 9 feet 9 inches in diameter by about 16 feet 10 inches long, with three corrugated furnaces of 36 inches diameter in each. Thickness of boiler shells seven-eighths inch. Total grate surface, 100 square feet; total heating surface, 3,630 square feet.

Forced draft will be by closed fire-room.

The following are the dimensions of the machinery of the *practice vessel*:

Cylinders 13½, 21, and 31 inches in diameter, by 20 inches stroke. I. H. P. 1,300 at 240 revolutions per minute and 160 pounds working pressure. Piston-valves for H. P. and I. P. cylinders 6 inches in diameter. Crank-shaft journals and pins 6 inches diameter with 3-inch axial holes. Cranks at 120°, and sections of shaft interchangeable.

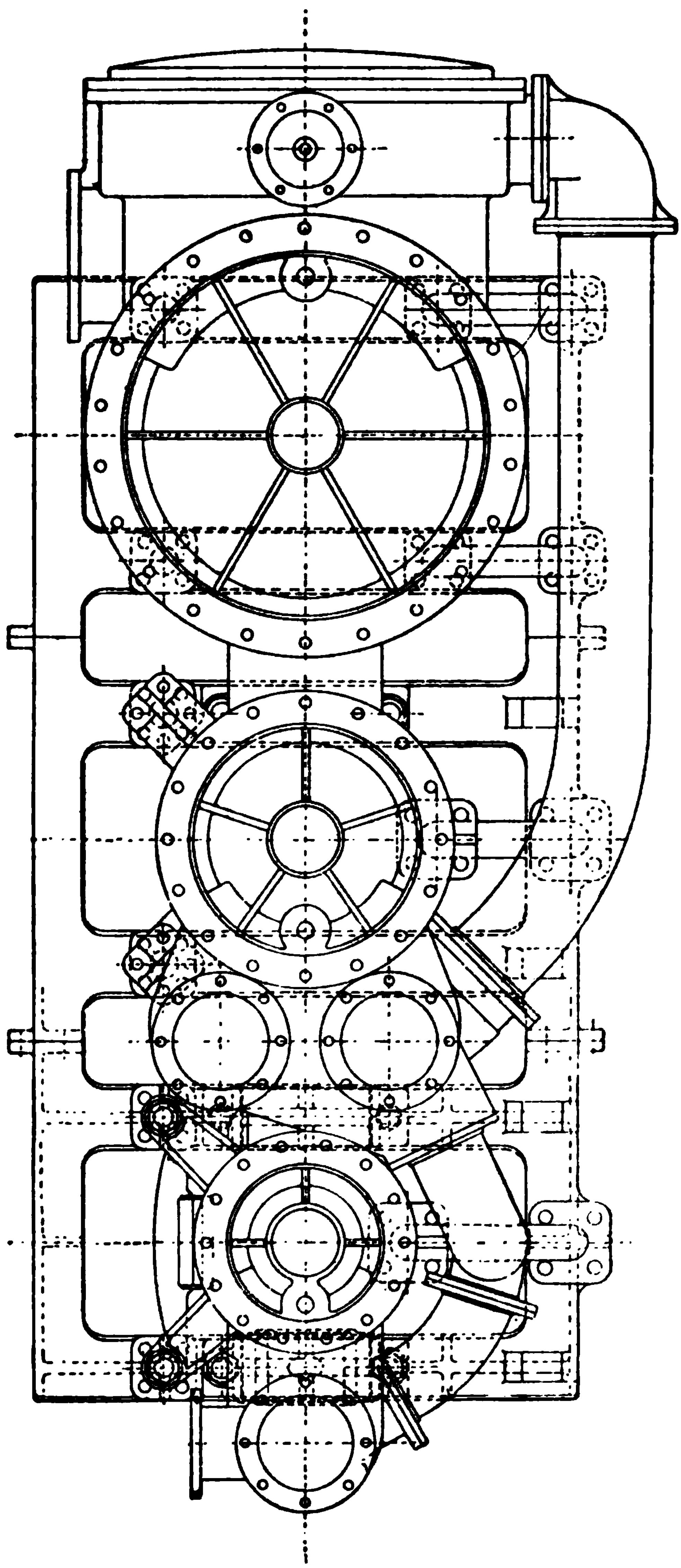

FIG. 74.—ENGINES OF NEW U. S. CRUISERS, 12 AND 13.

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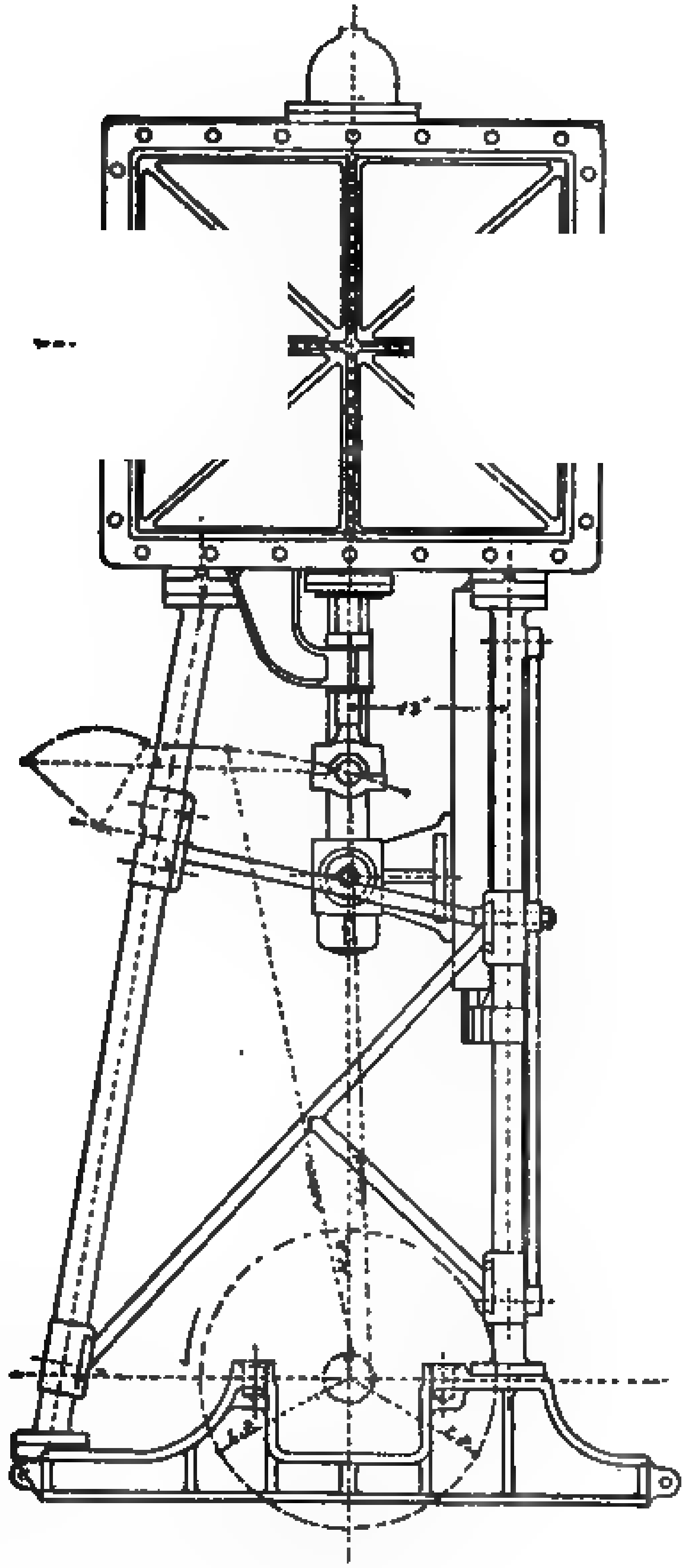

FIG. 75.—ENGINE OF U. S. CRUISERS 12 & 13.

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FIG. 76.—ENGINES OF NEW U. S. CRUISERS, 12 AND 13.

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Cooling surface of main condenser 1,820 square feet. Diameter of air-pumps 12 inches, stroke 12 inches. Capacity of circulating pump 2,600 gallons per minute.

There will be two boilers 8 feet 8 inches diameter and 17 feet long, with two corrugated furnaces of 39 inches diameter in each. Total grate surface, 78 square feet; total heating surface about 2,640 square feet; thickness of shells, three-fourths inch. The forced draft will be by closed ash-pits.

CHAPTER XXVII.

TRIPLE-EXPANSION SCREW-ENGINES AND BOILERS OF THE U. S. S. PHILADELPHIA, DESIGNED AND BUILT BY THE WM. CRAMP & SONS SHIP AND ENGINE BUILDING CO., PHILADELPHIA.

GENERAL DESCRIPTION.

THE engines have, each, a high, intermediate and low-pressure cylinder, of 38, 56, and 86 inches diameter, respectively, and a piston-stroke of 40 inches.

The engines are placed in separate water-tight compartments, and are duplicates, the low-pressure cylinder being forward in the forward, and aft in the after compartment; the forward engine turning the port-propeller.

The main steam-valves are of the piston type; there is one for the high-pressure, two for the intermediate, and two for each low-pressure cylinder, worked by radial valve-gear and arranged for a minimum cut-off of 0.4 of the stroke of the high pressure, and 0.5 in the intermediate and low-pressure cylinders.

Each piston has one piston-rod secured to a cross-head, which runs on guides, supported by the engine bed-frame.

Each crank-shaft is built-up, made of steel, in one piece, with cranks at equal angles and with the necessary coupling discs forged on.

The castings containing the crank-shaft bearings are cast in five pieces for each engine. They are bolted to engine keelsons, and stayed to cylinders by steel tie-rods and the engine bed-frames.

Each main engine has two air and two bilge-pumps, all vertical and single-acting, driven by an independent compound engine. Also one centrifugal circulating pump arranged with bilge as well as sea-injection. Air-pumps deliver into one feed-tank of boiler-iron. The tank has a capacity of about 500 gallons, and is partitioned and fitted as a filter.

The shells of condensers are cylindrical and made of brass. They are fitted with brass tubes $\frac{5}{8}$ inch diameter outside, and have a total cooling surface of about 13,500 square feet measured on the outside of the tubes. The tubes are placed fore-and-aft, the water circulating through them and thence overboard through outboard-delivery valves.

Suitable baffling and supporting plates are arranged in each condenser to assist in the circulation of steam and to support the tubes.

There are two vertical duplex pumps fitted in each fire-room of ample capacity for feeding the boilers. One set in each fire-room is fitted to

draw from feed-water tank and bottom of forward condenser and discharge through boiler check-valves. The second set in each fire-room is fitted to draw water from tank, sea, bilge, and boilers, and discharges into fire-main, through boiler-checks, and overboard. The pump in forward fire-room likewise discharges through distiller.

A horizontal pump is placed in each engine-room, and fitted to draw water from the sea and bilge, and to discharge into fire-main and through outboard-delivery.

The water-cylinders and chests of all pumps are of composition.

The distilling apparatus consists of one distiller capable of furnishing 3,000 gallons of potable water in twenty-four hours. The circulating water for the distiller is supplied by the auxiliary pump in forward fire-room.

The propellers are three-bladed, right and left-handed, respectively, of about 14 feet 6 inches diameter, and are made of manganese bronze.

There are four double-ended horizontal return tubular boilers, containing an aggregate grate surface of 624 square feet, arranged fore and aft, in two water-tight compartments, with four athwartship fire-rooms. Each boiler is 14 feet in diameter and about 20 feet long.

The smoke-pipe is fixed, and its top is about 60 feet above the grates.

Fire-rooms are arranged to work under air-pressure when required, and are fitted, each, with two blowers, capable of supplying continuously, with ease, sufficient air for the required horse-power.

DETAILED DESCRIPTION.

Cylinder Casings.—The cylinder casings, which include the steam and exhaust ports and passages, inboard heads, and valve-chests, are of cast-iron.

They are fitted with cylinder linings of hard cast-iron.

The cylinder casings have extensions cast on them with flanges 1¾ inches thick for securing them to engine bed-frames.

The cylinder casings and covers are covered with non-conducting material, and neatly lagged with black walnut.

Receivers.—The receivers consist of the spaces between high-pressure and intermediate, and intermediate and low-pressure piston-valves and their connecting pipes.

There is a 3½-inch copper pipe, with composition stop-valve, connecting main steam-pipe to each receiver space; and a composition safety-valve, with nickel seat of 2¾ inches diameter, on each receiver, the intermediate pressure weighted to 80 pounds per square inch above the atmosphere, and the low-pressure to 30 pounds.

Cylinder linings.—The linings are of cast-iron as hard as tools can work, with central ring or rest. They are 1⅛ inches thick, and are accurately fitted and secured to the casings. They are smoothly and accurately bored to diameters of 38, 56, and 86 inches respectively, and for a piston-stroke of 40 inches. Cylinder linings are jacketed.

Cylinder Heads and Covers.—The cylinder-heads are cast solid with cylinder casings and amply stiffened by ribs. They have suitable openings for the stuffing-boxes. The cylinder covers are made of cast-steel and well-ribbed. Each low-pressure cover has a man-hole cast in, which is faced to receive the man-hole plate. The cylinder covers are faced true on the inside. They have faced flanges, and are secured to cylinder casings by wrought-iron bolts with finished nuts. Bolts are spaced not over 6 inches apart.

Holding-down Bolts.—All holding-down bolts for securing the engines in the ship are fitted with locked nuts.

Man-holes and plates.—The man-holes in low-pressure cylinder covers are 20 inches in diameter. The plates are turned to loosely fit the holes, faced on the inner surface to fit the facing-strip on cover or head, and finished on the outside.

Valve-Chests and Covers.—The valve-chests

have openings at each end for inserting and removing the valves, and are closed by single-plate covers of cast-iron, well ribbed, finished on outside with faced flanges 3¼ inches wide and 1¾ inches thick.

The inboard covers contain the valve-stem stuffing-boxes; the outboard serves as the valve-stem guide. The packing spaces are fitted with metallic packing. The covers are secured in place by 1-inch bolts, spaced 6 inches apart, and with finished wrought-iron nuts. Suitable bosses are cast on the upper surface of steam-chests, directly over each steam-port, for the attachment of oil-cups.

Valve Liners.—The valve liners are made of cast-iron of the toughest quality, combined with a suitable degree of hardness. They are ⅞ inch thick, accurately bored and turned; then forced into seats after the ports were cut out 3½ inches for H. P. and M. P., and 3⅝ inches for L. P. The bridges in ports are 1 5/16 inches wide.

Main Steam Piston-Valves.—The piston-valve is made of composition, 7/16 inch thick in the body of the valve.

Each end of all valves is made steam-tight by two packing-rings of composition, ⅝ inch square in cross section, cut obliquely, and held in place by a composition follower and wrought-iron bolts. The distance-pieces for separating the packing-rings at each end of the valves are made of composition.

Main Valve-Stems.—The valve-stems are made of steel, 2⅜ inches diameter where they pass through the valves, and 3½ inches diameter in the stuffing-boxes.

Throttle-Valves.—The main steam-throttle for each high-pressure cylinder consists of a disc-valve 15½ inches diameter of opening, opening with the pressure in the steam-pipe, and is operated by suitable gear with hand-lever adjacent to hand-lever of steam reversing-gear.

Valve-Gear.—The valve-gear is of the radial type. The cut-offs of all cylinders are capable of being adjusted independently of each other.

The distribution of steam in backward gear is such as to permit the engines to be reversed quickly and to run astern at full power.

The eccentrics are made of cast-iron. Each eccentric is made in two parts, securely fastened together by two mild-steel bolts. They are truly bored to fit the shaft and properly secured to the same. They are truly turned to a suitable eccentricity, and recessed at the sides to fit the flanges of the eccentric-straps.

Each eccentric-strap is in two parts, of cast steel, with white-metal linings. The two parts are firmly fastened together by two mild-steel bolts with lock nuts. The two parts of the strap are separated by suitable brass chipping-pieces. A prolongation of one part of each eccentric-strap forms the eccentric-lever.

Each eccentric-lever has two mild-steel pins, one with a hardened steel thimble securely fastened.

One of these pins engages with the radius-link and the other with the valve-connecting-rod.

The movement of each valve is regulated by a reversing-arm and a radius-link.

Each reversing-arm is carried in bearings rising from the top of the corresponding crank-shaft bearing, with its main centre line parallel to the axis of the crank-shaft, and in the same vertical plane. The reversing-arm with its journals is of cast-steel. A forged steel pin is secured in the free end of the arm to engage with the radius-link. Each radius-link engages at one end with this pin and at the other with the lower pin on the eccentric-lever.

Each valve-connecting-rod engages at one end with the corresponding pin in the eccentric-lever and at the other end with a pin in an arm on the valve-motion rock-shaft.

The valve-motion rock-shafts are carried in bearings bolted to the cylinder casings, and have arms set at suitable inclinations to each other by which the motion is transmitted to the valve-stems by links.

The radius-links, valve-connecting-rods and valve-links are forged of mild steel, finished all over.

All joint-pins are of steel.

All working bearings are of phosphor-bronze or other composition.

The reversing-arm bearings are cast on the main pillow-blocks. The valve-motion rock-shafts and arms and reversing-arms are of cast-steel.

The radius-links are capable of adjustment so as to preserve a constant distance between centers when taking up lost motion.

Fixed trammels are furnished, suitably protected from injury, for setting the radius-link centers to their proper distances.

The valve-stems are marked and furnished with fixed trammels for setting the valves without removing the valve-chest bonnets.

A spare set of bearings is furnished for all adjustible joints.

All parts of the valve-gear are suitably marked for convenience of putting together properly when overhauling.

Reversing-gear.—Each engine has a steam reversing-gear with the cylinder placed vertically. Each main engine has one hand-reversing lever, which is conveniently placed to be worked from the working-platform. The reversing-lever sectors have adjustable stops to prevent the hand-levers being thrown beyond the full-ahead and astern positions.

The reversing engines exhaust into the respective condensers.

Steam Governor.—There is an efficient governor of an approved type, with all necessary connections fitting to the reversing lever of each engine, so as to control the admission of steam to each cylinder for preventing racing in rough weather.

Cylinder Relief Valves.—There is an automatic relief valve of 3 inches diameter, located at each end of each cylinder; these valves are guided by loosely fitting wings. They are kept on their seats by the pressure of steam in their respective receivers and by a light spiral spring.

Cylinder Drain Valves.—There is fitted to each end of each cylinder, a drain valve of approved design, with 1-inch opening. These valves are made of composition and are flanged and bolted to bosses on cylinder casings. They are arranged to work by hand-levers at working platform if found necessary.

Pistons.—The pistons are made of steel, the thickness of metal 2¾ inches at center and 1½ inches at periphery; that around the eyes of the piston-rods is 2¾ inches. Each piston has one cast iron wearing-shoe upon which it rests. These shoes are so fitted that they can be adjusted to take the wear. The packing-rings are ¾ inch thick, ⅝ inch wide, and are adjusted by steel springs of proper tension.

Piston-Rods.—The piston-rods are of steel, finished 7½ inches diameter, fitted and secured

to the pistons by iron nuts. The piston-rods are fitted into cross-heads and secured by nuts.

Cylinder Tie-Rods.—The tie-rods securing the cylinders to pillow-block frames are made of steel, turned to a diameter of 4¾ inches. They have T-heads forged on each end. These rods are secured to the pillow-blocks and to the cylinders by steel bolts 2¾ inches diameter.

Piston-Rod Stuffing-Boxes.—The piston-rod stuffing-boxes are formed in the cylinder-heads, and are fitted with cast-iron bushings and glands. They are fitted with an approved metallic packing.

Cross-Heads.—The cross-heads are of steel, finished all over, and fitted with cast-steel slippers, lined with white metal, 20 inches wide, 24 inches long. The connecting-rod journals are 9 inches diameter and 10 inches long.

Engine Bed-Frames and Cross-Head Slides.—The engine bed-frames and cross-head slides are of cast-steel, well secured to cylinders at one end and to the bed-plates at the other.

Connecting-Rods.—The connecting-rods are of steel, finished all over.

They are 86 inches long between centers, 7 inches diameter of neck at crank-pin end, and 6 inches diameter of neck at cross-head end. The crank-pin and cross-head boxes are made of composition. The crank-pin boxes are secured to rod by two 4¾-inch steel bolts, and each

cross-head box by two 3¾-inch bolts; the nuts are secured by proper set-screws. The boxes for crank-pins are 2½ inches thick, and for cross-heads 1¾ inches thick, accurately fitted to pins and rods.

Crank-shafts.—The crank-shaft for each set of engines is made of steel, built up with solid webs and couplings with cranks at equal angles. The shaft-journals are 14½ inches diameter and have a total length of about 122 inches.

The crank-webs are 9½ and 10½ inches thick. The crank-pins are 15 inches diameter and 17 inches long.

The couplings are 3¾ inches thick and 28½ inches diameter. The after, middle and forward crank-pins and shaft-journals have holes axially through them, respectively 7, 8 and 9 inches diameter.

Crank-shaft Boxes and Caps—Boxes of composition are fitted to the main pedestals. The caps are of steel and both boxes and caps are made with recesses for white metal linings.

Each cap has a hole through it of sufficient size to feel the journal. Each cap is secured by two 4-inch steel bolts, and with set-screws to prevent nuts from working loose.

Composition boxes are hollow for water circulation. Both shaft and crank-pin brasses are scraped to accurately fit their journals.

Bed-plates and Pillow-blocks.—The bed-plates

for pillow-blocks are made of steel in five castings, from which spring the pedestals for crank-shaft bearings. The plates and pedestals are cast hollow, with walls 1⅛ and ⅞ inch thick, the metal around the boxes is 1¾ and 2 inches thick. The bottom flanges of bed-plates are 1⅛ inches thick and 3½ inches wide.

Surface Condensers.—The condenser-chests are cylindrical in form, "built up" of sheet-brass ¼ inch thick, amply sustained by angle and T-rings and composition flanges for the tube-plates.

The exhaust and discharge-nozzles, also the chambers for the circulating waters and the covers for the same, are of composition as thin and light as practicable, combined with ample strength and stiffness. The diameter of the exhaust opening is 30 inches, and of the discharge openings to air-pumps 16 inches. The injection and delivery openings for the circulating water are 15 inches in diameter. All flanges 2½ inches wide.

A 1 inch salt-water feed-valve is attached to each condenser.

Each chest contains 3,440 seamless-drawn brass tubes, ⅝ inch outside diameter, of No. 20 B. W. G. thickness, spaced 15/16 of an inch between centres.

The exposed condensing length of tubes is 12 feet, having a total cooling surface of 6,750

square feet. The tubes are thoroughly tinned inside and out.

The tube-plates are of brass, $\frac{7}{8}$ inch thick, stayed to heads by eight 1-inch stays, bored for the tubes, and counter-bored $\frac{7}{8}$ inch diameter and $\frac{9}{16}$ inch deep, the packing is compressed by composition glands screwed into the plates, and having a device for preventing crawling of the tubes.

The tubes are suitably supported by an approved system of composition diaphragm and deflecting plates in each condenser.

The condensers are located behind the engine cylinders and well secured in the ship.

Additionally, the condenser in the forward compartment is fitted with straightway-valves in its cylinder exhaust-pipe, and discharge-pipe close to air-pump, for closing all communication with the main engines when the condenser is used for auxiliary purposes.

Auxiliary Exhaust-Main.—The auxiliary exhaust-main, where it passes through the engine compartments, has a diameter of 6 inches, and is made of copper. It has in each engine compartment two exhaust connections—one to condenser and one to low-pressure receiver, each 4 inches internal diameter. All flanges are made of composition, faced not less than 2½ inches wide.

One 4-inch escape to atmosphere.

Air and Bilge-pumps.—Each engine has two

vertical single-acting air-pumps 24 inches diameter and 18 inches stroke, and two bilge-pumps 4 inches diameter and 18 inches stroke, operated direct by a special compound engine.

The pump-cylinders, valve-chests, covers, bonnets, buckets, bucket-rods, plungers, valve-seats and guards are made of composition; the air-pump valves of hard rubber. The bilge-pump valves of rubber, with two layers of canvas. The pump-cylinders are $\frac{1}{16}$ inch thick. All possible conveniences are attached to air-pumps for examination of valves.

Centrifugal Circulating-pump.—The centrifugal circulating-pump in each engine compartment is of approved design, with 15-inch suction and discharge openings. All pipes are fitted with composition flanges of $2\frac{1}{2}$ inches face.

The pump is operated by a special engine connected directly with it; takes steam from both main and auxiliary steam-pipes direct, and exhausts into the condenser through the auxiliary exhaust-main.

Pump-casing, fan and pump-shaft are of composition.

Injection-valves.—The chests, valves, seats, bonnets, glands, screw-stems and hand-wheels of the injection-valves are of composition.

Each valve has an opening through the seat of 15 inches diameter.

Each chest has a nozzle of $3\frac{1}{2}$ inches dia-

meter of opening under the valve for fire-pump suction.

Composition strainers equivalent in area to twice the area of valve cover the openings through the ship.

Bilge-Injection.—A copper pipe of 12 inches internal diameter connects each main injection-valve chest with the bilge in its engine compartment. Each pipe has attached to it a composition non-return valve of 12 inches diameter of opening.

Outboard-Delivery Valves.—The chests, bonnets, seats, valves, stems and glands of outboard-delivery valves are of composition.

The valves are fitted as checks, to open by pressure from inside, to cover openings through seat 15 inches diameter. Each chest has a valve of 3½ inches diameter of opening outside the main valve for the bilge discharge from fire-pumps, and one for the discharge from main bilge-pumps.

Sea-Valves.—There are two sea-valves of not less than 3½ inches diameter of opening in each fire-room, one to be used for blow and the other for sea-suction.

The chests are provided with suitable nozzles for connecting them with pipes, leading to boilers and pumps.

The chests, bonnets, valves, seats, stems, glands and hand-wheels are made of composition.

The suctions have composition strainers, with holes through them of an aggregate area not less than twice the area of valve openings.

Feed and Auxiliary-Pumps.—There are two vertical duplex-pumps of approved design, located in each fire-room; each pump has water-cylinders of 5 inches diameter and a piston-stroke of 12 inches. One pump in each fire-room is connected to feed-tanks, bottom of forward condenser, and boiler-checks only, and has a screw check-valve on both suction and delivery-pipes close to the pump. The other pump in each fire-room is fitted to draw from feed-tank, sea, bilge and boilers; and to deliver water into any of the boilers by a distinct set of feed-pipes and check-vaves independent of the main feed system, and, likewise, into the fire-main and overboard.

In addition, the forward auxiliary pump is fitted with a suitable discharge-pipe for flushing the head and for distilling purposes. It discharges its water overboard through a sea-valve forward.

Fire and Bilge-Pumps.—There is a vertical steam-pump of approved design placed in each engine-room. They are fitted with the requisite valves and connections for use as bilge and fire-pumps, and draw water from the sea through a valve or main injection-chest or through bottom of ship and from the bilge. They deliver

overboard through valves on outboard-delivery valve-chests, and into fire-main.

Each pump has a steam-cylinder of 8 inches diameter, water-cylinder of 5 inches diameter, and a stroke of 12 inches.

Distiller and pump.—The distilling apparatus is located on the berth-deck, and consists of one distiller of approved design capable of furnishing 3,000 gallons of potable water in 24 hours. It takes its steam from the auxiliary boiler by an independent stop-valve and pipe.

The auxiliary pump in fire-room is used as a circulating-pump for the distiller. One independent pump for distiller.

The circulating water, after passing through the distiller, goes forward through a proper copper pipe for use in flushing the heads; a copper bye-pass pipe, fitted with suitable valves, connects the discharge of the pump used to circulate water through the distiller with the pipe leading forward to the head, for use when from any cause the distiller is shut off.

The distiller is fitted with a filter and with the pipes necessary for running the distilled water into the fresh-water tanks.

Pump-cylinders.—The water-cylinders of all steam-pumps are made of composition.

All pumps have screw check-valves in section and delivery-pipes close to pump-chambers, and stop-valves in both steam and exhaust-pipes.

All suction-pipes leading to bilge, excepting those from the circulating-pumps, are fitted with Macomb bilge-strainers. The steam-cylinders of all pumps, blowers, and other auxiliary machinery have their exhaust-nozzles connected to an exhaust-main, which will pass through engine and fire-rooms. This main is connected to both main condensers and to the second receivers of both engines, and has a discharge into the atmosphere, and is furnished with the necessary valves for governing the direction of the exhaust. Additionally, the main feed-pumps are supplied with means of turning their exhaust steam into their feed suction-pipes.

Working-platforms. — Working-platforms of wrought-iron are situated below the center of shaft and on each side of the bulkhead, between the engines, convenient to which are arranged all the handles, levers, and connections for operating the engines, with the counters, revolution-indicators, clocks, steam and vacuum-gauges in plain view.

Ladders are provided as means of escape from engine-rooms when the water-tight doors are closed, and are located on the bulkhead separating the engine compartments.

The engine-room stairway for ordinary use is accessible from the berth-deck through a door in engine-room hatch bulkhead, and has its landing on the working-platform in the forward

engine compartment. A door near this stairway communicates with after engine compartment, and suitable footways are arranged for access to the moving parts of the machinery, fitted, where required, with brass hand-rails and finished wrought-iron stanchions.

Feed-water Tanks.—A feed-water tank is placed in forward engine-room. It is made of wrought-iron $\frac{5}{16}$ inch in thickness, and has a capacity of about 500 gallons, and is fitted as a filter and is provided with a vapor pipe, a float-valve for preventing access of air to feed-pump, an overflow-pipe and a glass gauge.

A supply-pipe leads from this tank to the main feed-pumps in fire-rooms.

Line-Shafting.—The line and thrust-shafting of both engines are made of steel, 13¾ inches diameter, with a 6-inch hole axially through it, and are supported by spring bearings where necessary. The thrust-shafts have eleven raised collars 1⅜ inches thick, 18 inches outside diameter, and 1⅞ inches space between them.

Propeller Shafting.—The propeller-shaft of each engine is made of steel in two lengths. The forward length is 14½ inches outside diameter, with 8-inch hole axially through it. The after length is 15 inches outside diameter, with 9-inch hole axially through it. The hole is reduced at propeller end to correspond with diameters. The forward length is cased with brass

in stern-pipe. The after length is covered at bearing only. Sleeves at bearings ⅞ inch thick, remainder ½ inch thick.

Screw-Propellers.—The propellers are made of manganese-bronze, 14½ feet diameter; have adjustable blades and turn outward in forward motion.

Outside and Stern-Pipe Bearings.—The stern-pipes and outside bearings have composition bushings fitted with lignum-vitæ staves, with the proper flanges for securing them in position.

The bearings in stuffing-boxes are 24 inches long, the outer ones 32 inches and those in hangers 48 inches.

All lignum-vitæ bears on end of grain.

Stern-Pipe Stuffing-Boxes. — The stuffing-boxes are made of composition, with a packing space 1 inch wide and 6 inches deep, fitted with followers made in two parts with a space of 1½ inches between them, and secured in place by Tobin's-metal bolts.

Thrust-Blocks and Bearings.—The thrust-blocks and caps are of cast-iron, lined with white metal, and made for a circulation of water through them. They are provided with stuffing-boxes and glands at both ends for retaining the oil.

The caps are made with lugs locking into the blocks, and have ample oil and grease-cups with hinged covers.

Each cap is well secured in place by four wrought-iron bolts 1½ inches diameter.

The blocks rest on sole-plates riveted to the foundations built in the ship, to which they are secured by twelve bolts 1¼ inches diameter, and fitted with keys so that they can be accurately adjusted to line in any direction.

Spring-Bearings.—Spring-bearings are 18 inches long.

Turning Gear.—Steam-gear is provided for turning the main engines.

Water-pipes.—Seamless brass water-pipes 2 inches diameter are fitted with the necessary valves in each engine-room. The water supplied from a pump.

They have two branches of 1¼ inches diameter to each main and crank-pin bearing, one branch of 1 inch diameter to each eccentric, and two branches 1¼ inches diameter to each thrust-bearing.

Also two of ¾ inch diameter, screwed into each crank-shaft pillow-block, with holes leading down through brasses to tops of journals, and one of ¾ inch diameter screwed into each cross-head slide.

All branch water-pipes are fitted with valves for regulating the supply of water to bearings.

Journal-boxes.—All journals or moving parts of iron or steel are run in boxes either of composition or of cast-iron or steel lined with white-metal.

The crank-pin and crank-shaft boxes are lined with Parson's white-metal.

Indicator Fittings and Motions.–Indicator connections for each end of each steam-cylinder are fitted, as near as possible, to the bores of their cylinders, and so located as to be easily accessible.

The indicator-motions are so designed as to give the indicator-barrels motions coincident with those of the pistons, and of sufficient throw to give a diagram of 4 inches length.

Revolution-indicators.—Revolution-indicators, showing on suitable dials the speed and direction of the engines, are placed in each engine-room, and suitable dials for showing in which direction the engines are turning are placed on deck.

Oil-Cups.—Each crank-pin is fitted with a telescopic or wiping arrangement, of approved design. All crank-shaft bearings have ample oil-cups with hinged covers, tube and wick-holders, and so arranged that the amount of oil passing down each tube to the journals can be seen and regulated. Wipers carried by the upper ends of the eccentric-leavers furnish oil for lubricating the eccentrics and all connections of the eccentric-levers. These wipers to take oil from strips of webbing supplied by oil-cups suitably supported and capable of adjustment so as to feed oil in all positions of the valve-gear, and also so arranged as to make the supply of oil to the various parts independently adjustable.

All other joints or moving parts not otherwise referred to, and especially the cross-head slides and the valve connections, have finished brass automatic oiling-gear of approved design, capable of supplying sufficient lubrication while the engines are in operation without waste of oil.

All oil-cups are such as can be easily filled while the engines are running at maximum speed, and have an oil capacity for at least four hours running.

All fixed bearings have drip-cups cast on where possible, of cast-brass and properly fitted.

All such cups have drain-pipes, and cocks of at least ½ inch diameter, which can be used while the engines are in operation.

All moving bearings have drip-cups or pans of sheet brass where necessary.

Holes Through Ship.—All holes through the ship are covered by valves on the inside, and fitted with zinc protecting-rings.

Pump Connections to Fire-Main.—The fire and auxiliary feed-pumps each have a discharge-pipe, with straight-way stop-valve, connecting it with the fire-main running fore and aft, and a branch from each discharge-pipe near the pump is fitted with standard hose-connections and straightway-valve.

Eye-Bolts.—Wrought-iron eye-bolts and traveler-bars are properly located and secured wherever required for lifting different parts of ma-

chinery, and particularly the covers of cylinders and valve-chests, the covers of air and circulating-pumps and their valve-chests, the condenser-bonnets, the connecting-rods, the caps of pillow-blocks of crank-shaft and line-shaft journals and of thrust-bearings.

Securing Engines in Ship.—The engines are adjusted and lined upon the engine-keelsons, and accurately in line; the spaces around holding-down bolts between sole-plates and keelsons are filled by accurately fitting wrought-iron washers, upon which the holding-down bolts are set up and locked in place.

Drain-Pipes and Traps. — All receptacles where condensed steam is likely to accumulate are provided with drain-pipes and cocks of ample capacity leading to automatic traps (fitted with bye-pass pipes and valves), which discharge into feed-tanks.

The drain-pipes from safety-valves are connected ½ inch below the level of valve-seats.

Boilers and Attachments.—There are four double-ended horizontal return tubular boilers, constructed of open-hearth steel. The boilers carry a working pressure of 160 pounds by gauge.

All plates are planed on their edges, and thoroughly calked inside and out wherever accessible. Butt-jointed seams are covered with straps, and all rivet-holes are drilled to full size. Each

boiler is 14 feet mean diameter outside, and 20 feet long, and has eight furnaces 36 inches least internal diameter.

Grate-surface.—Total grate-surface in four boilers is 624 square feet.

Grate-bars.—The grate-bars are of wrought-iron, in two lengths, of approved pattern. The furnace-fronts, bridge-walls, and bearers are properly fitted to support the bars.

Tubes.—Each boiler contains nine hundred and sixty-four lap-welded wrought-iron tubes. Every third tube, vertically and horizontally, is a stay-tube, and is No. 8 B. W. G. thick and 2¼ inches external diameter. The other tubes are No. 12 B. W. G. in thickness, 2¼ inches external diameter.

The stay-tubes are screwed into both heads, the ends at front heads to be swelled. They are expanded into both heads and beaded over at combustion-chamber ends.

Boiler-shells.—The shells are of plates 1⅛ inches thick, the longitudinal joints are double-strapped and double-riveted each side of seams. The circular joints are lap-jointed and double-riveted.

Tube-sheets.—The tube-sheets are $\frac{9}{16}$ inch in thickness, and are accurately drilled for the tubes.

Boiler-heads and Braces.—The upper portion of the heads are ⅞ inch thick and the lower

portion $\frac{9}{16}$ inch thick. The upper portions of heads are braced by three rows of steel stays 2½ inches diameter in the body, 17 inches between centers horizontally, and 13 inches vertically. Jaw-braces are of iron. The tops of combustion-chambers are stayed to shell by 1¾-inch iron braces with crowfeet on top of the chambers, placed 6¼ inches apart in length of boiler and 13 inches in the diameter. All steel braces are without welds in length or eyes. The through braces are made with nuts on both sides of boiler heads, having raised threads on ends.

Furnaces.—The furnaces are of the best steel, welded at joints, and corrugated. They are 36 inches diameter at the inside of corrugation, ½ inch thick, and are single-riveted at their junction with front heads and combustion-chambers. Ash pans of ¼ inch wrought-iron are fitted in all furnace-flues reaching from front to bridge-wall.

Bridge-Walls.—A bridge-wall of approved pattern is fitted in each furnace. The upper part is finished with fire-brick. The bridge-walls are easily removable.

Combustion-Chambers. — The combustion-chambers are 54 inches deep; the sides, tops, ends, furnace-plates and tube-sheets $\frac{9}{16}$ inch thick. The sides stayed by steel screw stay-bolts 1¼ inches diameter, spaced 6⅝ inches from center to center.

Smoke-Boxes and Uptakes.—The smoke-boxes and uptakes are made with single shell, covered with an approved non-conducting substance and protected by an outside shell. The inner shell is secured to boilers by 2½-inch angle-irons. Both shells are made of iron $\frac{3}{16}$ inch and ⅛ inch thlck, respectively.

The connection doors are made of wrought-iron with double shells, and fitted with hinges and catches of wrought-iron. The outside shell is $\frac{3}{16}$ inch thick, and the lining ⅛ inch thick.

Furnace fronts. — The furnace-fronts are of wrought-iron ⅜ inch thick, with cast-iron perforated liners.

Furnace-doors. — The furnace-doors are of wrought-iron ¼ inch thick, with cast-iron perforated liners.

Ash-pit Doors.—The ash-pit doors are of wrought-iron ⅛ inch thick, flanged 1 inch deep, and fitted to place so as to thoroughly close the ash-pits, and fit lugs on bulkhead when not in use.

Saddles.—Each boiler is to rest on three saddles, which are built in and form part of hull. The boilers are secured by double angle-irons riveted to saddles and bolted to boilers.

Smoke pipes.—There are two smoke-pipes 60 feet in total height above the upper grates. They are 6 feet 9 inches in diameter, made of wrought-iron plates; the lower course No. 7

B. W. G. thickness, the upper ones No. 8. The pipe is stiffened by flat bands on the inside, at top and bottom, and by a band 4 inches wide and 1 inch thick on outside at top. The pipe is inclosed its entire length by a jacket, leaving an annular space of at least 3 inches. The jacket is of wrought-iron, No. 13 B. W. G. thick, and covered by a hood for the escape of hot air. The pipe and jacket are made with strapped butt-joints. The smoke-box and breeching inclosed by a jacket made of No. 16 iron. The pipe is provided with stays, eyes and shackles, and is supported in such a manner as to relieve the up-takes of its weight. A pivoted damper is fitted in each smoke-pipe.

Dry pipes. — Each boiler has a perforated tinned-brass dry pipe of reduced diameter at the internal end, but of same diameter at front end as the steam-pipe with which it is connected. It is placed as high as possible, and extends nearly the length of the boiler.

Its upper surface is pierced with holes 3/8 inch in diameter, spaced equidistant, their aggregate area to be twice that of the cross-section of the pipe.

Boiler Clothing.—The boiler shells and fronts are covered with approved material, which is protected by a galvanized-iron covering, the joints of which are lapped and bolted.

Safety-valves.—Each boiler has two automatic

spring safety-valves 6 inches in diameter, adapted to a maximum pressure of 160 pounds per gauge, and fitted with proper levers and approved mechanism for working them from the fire-rooms. The chests, valves and stems are of composition, and seats of nickel.

The chests are bolted to stop-valve chambers, and connected by copper pipes to the escape-pipes, which are also of copper. The seats of all safety-valves are at least ½ inch above the bottom of their chests.

Sentinel-valves.—There is a sentinel-valve of ½ square inch area attached to the front of each boiler, fitted with movable weight and notched lever, and weighted to close tightly against a boiler pressure of 175 pounds per square inch.

Water-gauges.—Each boiler has two composition water-gauges carrying glasses 16 inches in exposed length, and with outside pipe-connections to top and bottom of boiler, the bottom of glass being 1 inch below the highest heating surface. The water-level is marked on brass plate on outside of boiler.

There are four gauge-cocks on each end of boiler, placed 4 inches apart, the lowest cock 4 inches below the highest heating surface of the boiler.

Salinometer-pots.—There is a salinometer-pot of approved pattern for each boiler, fitted in an accessible position and suitably connected.

Auxiliary boiler.—There is one cylindrical horizontal return-tubular boiler for auxiliary purposes in after fire-room. It is made of open-hearth steel for a working pressure of 160 pounds by gauge. The dimensions are as follows: 8 feet 6 inches diameter, 8 feet 3 inches long, with two corrugated furnaces 2 feet 8 inches inside diameter, and 5 feet 10 inches long and $\frac{7}{16}$ inch thick. Shell-plates in one length $\frac{11}{16}$ inch thick, with double-riveted double butt-straps. The end plates above tubes are $\frac{3}{4}$ inch thick and properly stayed. Remainder of plates $\frac{9}{16}$ inch thick. There are 88 tubes 3 inches outside diameter. Smoke-boxes and uptakes are similar to main boilers and smoke-pipe, and are carried into after main smoke-pipe.

Auxiliary Steam-pipes and Valves.—Each boiler stop-valve chamber has an auxiliary stop-valve 5 inches diameter, bolted to the nozzle on its side and under the main valve. These valves are connected by an auxiliary steam-pipe of 5 inches internal diameter, with suitable branches leading to the pumps, heaters, distiller and auxiliary engines.

A branch pipe with stop-valve connects main and auxiliary steam-pipes in each engine-room.

A steam-gauge in brass case, with $4\frac{1}{2}$ inch dial, is attached to the auxiliary steam-pipe in each engine-room and each fire-room.

Bleeder.—There is a copper pipe, with stop-

valve at each end, 5 inches in diameter, leading from the main steam-pipe to each condenser. One valve in each engine-room worked from working-platform.

Check-Valves.—Each boiler has two feed check-valves, 2½ inches in diameter, having outside screw-threads on their stems; chambers, valves and stems of composition.

All check-valves have internal pipes.

Blow-Valves.—Each boiler has a bottom blow-valve of composition 2½ inches in diameter; also a surface blow-valve 1½ inches in diameter. These valves are connected by suitable pipes to the sea-valves.

The bottom blow-valves have internal pipes leading toward the bottoms of boilers, the surface blow-valves have pipes leading to the centers of boilers, with openings about 1 inch above the highest heating surface.

Feed and Blow-Pipes.—The main feed-pipes are made of copper tubes, 2½ inches in internal diameter, and in sections not exceeding 12 feet in length. The blow-pipes are 2½ inches in diameter. The branches are of copper. All nozzles and flanges are of composition. The several sections are expanded into flanges, then turned over and brazed. All flanges are united by forged bolts and nuts of Tobin's metal.

Boiler Stop-Valves.—The stop-valves are non-return valves. Each boiler has a composition

stop-valve chamber. The valve is 11 inches in diameter, fitted with a screw-stem of composition, made to turn independently of the valve, and to work in a composition nut supported by wrought-iron studs screwed into the cover. The valve is operated by a composition hand-wheel, 16 inches in diameter. Separate provision is made for working all boiler stop-valves from above the protective-deck.

Main Steam-Pipes.—The steam-pipes at stop-valves are 11 inches in internal diameter. The forward pair of boilers are connected by a separate 15½-inch pipe to forward engines, the after pair by a 15½-inch pipe to after engines; these two pipes are connected by a 15½-inch pipe. These pipes are of copper, the several sections united to each other and to the separators and valve-chambers by composition flanges of suitable size and thickness, riveted on and brazed. Where these pipes pass through water-tight bulkheads, they are provided with approved provision for expansion.

The steam-pipes and flanges are covered with an approved non-conducting material, covered and protected by an approved water-tight covering; this covering is secured to bulkheads where the pipes pass through them.

Escape-pipes.—There are two escape-pipes of copper, one abaft the forward and one abaft the after smoke-pipe, extending to the top of the pipe

and secured to it. There is one auxiliary escape-pipe about 4 inches diameter, connected with the auxiliary exhaust-main.

Pipe-clothing.—All main and auxiliary steam-pipes, exhaust-pipes, the separator and all steam-valves, are clothed with an approved non-conducting material, covered with canvas in double thickness, well painted. The covering is secured to bulkheads where the pipes pass through them. The pipes are also covered with black-walnut lagging with brass bands. Pipes in fire-room have galvanized sheet-iron casings.

Pipes through Bulkheads.—All pipes where they pass through water-tight bulkheads are provided with stuffing-boxes, or made tight in other manner.

Boiler drain-cocks.—There is a drain-cock having 1½ inches diameter of opening, fitted to each end of each boiler.

Separators.—There is a separator in each main steam-pipe, fitted with an automatic trap, a drain-pipe and valve leading to feed-tank, to bilge and overboard, and a glass water-gauge on the side.

Floor-Plates.—The fire and engine-rooms and their passages are floored with wrought-iron plates having corrugations on the upper surface and proper ledges and drain-holes. They are of wrought-iron ¼ inch thick, and all easily removable.

Blowers.—The fire-rooms are supplied with air by means of blowers, two to each fire-room. Each blower is driven by its own engines direct, and is capable of supplying, with ease and certainty, sufficient air for efficient forced draught.

Each fire-room ventilator is so fitted that it can be easily closed from fire-room in case its blower is stopped.

Ventilators.—Two ventilators, each 24 inches in diameter, are fitted in each fire-room. They deliver air to the inlet of the blowers placed under them. They have movable hoods and are made of iron 1/8 inch thick. The gears for turning the hoods are of composition.

Four ventilators, 18 inches diameter, are fitted—two to each engine-room; they lead down the engine-room hatches; their cowls are worked from the engine-room. All ventilator-cowls are made of copper No. 12 B. W. G., unplanished.

Ash-Hoists.—There is an ash-hoist arranged in ventilators of each fire-room, and a means for closing them when an air-pressure is required in the fire-room.

There is an approved ash-hoisting engine for each fire-room, to hoist 150 pounds from fire-room floor to deck in five seconds with 60 pounds steam-pressure. They are fitted with all necessary connections, including whip, and with a suitable brake to control the drum.

From each ash-hoist, on the upper deck, permanent overhead rails, suitably supported, lead to the nearest ash-chute on each side of ship. Each of these is fitted with a traveler of approved design, with all necessary appliances for carrying the ash-buckets. At the top of each ash-chute a dumping-hopper is fitted, so arranged as to fold up out of the way when not in use. The ash-buckets are balanced dump-buckets, with gear complete. All of the ash-hoisting and dumping gear is such that the buckets have not to be lifted by hand. A speaking-tube leads from the top of each ash-hoist to fire-room.

Air-tight fire-rooms.—Supplementary bulkheads and ceilings of light galvanized iron are fitted in the fire-rooms for the purpose of reducing the capacity of the space to be put under air-pressure. The ceiling is made movable beneath hatches. The vertical portion is provided with openings where passage ways are required, with suitable means for closing them.

All permanent and temporary joints and seams are made perfectly air tight.

HYDROKINETER.

There is connected to each boiler a Wier's hydrokineter, for circulating water in the boiler while raising steam, proper connections being made to auxiliary steam-pipe.

CHAPTER XXVIII.

EXAMPLES OF RECENT ENGINES.

Triple Expansion Engines of the Ocean Tug Triton.

FIG. 77 shows the engines of the ocean tug "Triton," built by the Atlantic Works, East Boston, Mass., and owned by Capt. Fred Luckenbach, of New York. The vessel is a fine representative of a new and staunch type of tug, especially adapted for sea service, with a length of 130 ft. 10 in., beam 26 ft. 6 in., depth of hold 14 ft. 6 in., and draught 13 ft. 6 in., the hull being of white oak, copper fastened. The engines are of the inverted vertical triple expansion description, with a high pressure cylinder of 15½ in. diameter, intermediate pressure cylinder of 24 in. diameter, and low pressure cylinder of 40 in. diameter, and a thirty inch stroke. The cylinders are of hard-grained cast iron, with the valve faces separate and bolted on. The crossheads are of wrought iron, with journals forged on, and gibs of cast iron, babbitted. The connecting rods are of wrought iron, and the line shaft is of wrought iron, 8½ in. diameter. The piston rods are of mild steel,

FIG. 77.—TRIPLE EXPANSION ENGI

Engines for the Ocean Tug Triton.

Opposite Page [illegible].

3½ in. diameter. The surface condenser forms a part of the framing, and has 950 square feet of cooling surface. Each engine has an independent cut-off, the connection of links to eccentric rods and to valve stem being adjustable, so that each link may be adjusted independently, and a steam reversing gear is provided, operated by a lever in the engine room. The screw is of cast iron, 10 ft. in diameter. The boiler is of the Scotch flue type, 13 ft. 6 in. diameter and 11 ft. 3 in. long, and is built for a working pressure of 156 lb. per square inch. The machinery is all strongly built and well finished. There is no extra work for ornamentation, but every part has the appearance of solidity, and is evidently intended to give a high degree of efficiency. The indicated horse power on trial was 720.

The design of the engines and arrangement of the cylinders, the high pressure being independent from the intermediate, is the design of James T. Boyd, constructing engineer of the Atlantic Works.

WELLS PATENT BALANCED COMPOUND AND QUADRUPLE EXPANSION ENGINES.

Figs. 78 and 79 show a Wells engine with a "natural" balance in weight of the two pistons, and their connections, at all angles of the cranks and at all speeds, also a balance of steam pressures. Equal weight being attached to opposite

sides of the crank shaft moving in opposite directions (in the same plane), the thrust of one is perfectly counteracted by that of the other, producing a perfect equilibrium and preventing vibration. While the weight of the moving parts in a single acting, or on an unbalanced engine, retards the motion of the shaft, by this method it is accelerated. An equal weight being attached to opposite crank pins, revolving around the shaft in the same plane, acts the same as a fly wheel, in carrying it over the centres, producing a rotary motion. Practice proves that it is far less liable to stop on the centers, which is owing to equal weights of the pistons and their connections.

Steam is admitted simultaneously to the bottom of the H. P. and to the top of the L. P. cylinders, and vice versa. The force on one cylinder-head is counteracted by an equal force on the other. Hence there can be no strains transmitted to the frame, and thence to the main bearing boxes. The ascending steam force on the small piston is equaled by a descending steam force on the large piston, which transfers the fulcrum from the main boxes to the crank shaft, concentrating the whole force in the shaft for useful effect. The main bearing boxes being relieved of the steam pressures, and also the thrust of the rods, their life will be made almost unlimited, as they will only be subject to the

FIG. 78.

WELLS PATENT BALANCED COMPOUNI

FIG. 79.

POUND AND QUADRUPLE EXPANSION ENGINE

To Face Pa[illegible]

friction and wear due to the weight of the shaft, which prevents heating and the liability to get out of line. The following shows the relief afforded in an engine of medium power. For example, take an 18x36 with an initial pressure of 100 pounds. Main bearings are relieved of 17 tons of steam pressure at each stroke of the pistons, and also the thrust of the rods, at a velocity of 500 per minute, gives a total relief of 21 tons per stroke. It will readily be seen that the greater the power required the more important it is to obtain a balance. It will also be seen, by perfectly balancing the moving parts, as is done in this case, no power is lost in lifting the weight or forcing up the pistons. And for the same reason uniformity of motion is obtained, insuring great durability. Another very important feature of the principle is that the hull of the vessel is relieved of all strains (from the force applied) excepting the pressure from the screw. Quadrupled for high steam pressure, this engine occupies one-third less space than the ordinary triple. The important advantages to be derived by using steam upon the balance principle is practically unknown, for the reason that heretofore no engine has been constructed to obtain a balance that did not embrace grave mechanical objections. Hence there was no means of putting it to practical test, until this design was put in use. The principle demonstrates in

practice an important gain in power, great durability, uniformity of motion, and the absence of all vibration.

MODERN HIGH SPEED YACHT ENGINES.

The increasing interest now taken in steam yachting has necessitated a better class of machinery than heretofore. Speed is now of the first importance, though most owners are particular about the appearance of their engines and wish them to be as attractive as possible. Fig. 80 represents a triple expansion engine built by John W. Sullivan, New York, and is a fair example of the high grade of work turned out by this shop. The cylinders are all three in one casting so as to save weight, the back of columns, which carry the guides for the crossheads, are all cast hollow of steel, and the bed plate is also of steel for the same reason. Piston valves are used throughout driven by Bremme gear, so that the engine is very open in appearance, easy of access and greatly simplified. Light steel columns support the front of the cylinders, which being polished add greatly to the appearance of the engine. The crank shaft and eccentrics are forged steel in one piece, while the connecting and piston rods, valve gear and all bolts and nuts, are also steel. The connecting rods are bored through the centre in order to further reduce the weight. The air and two feed pumps

FIG. 80.—TRIPLE EXPANSION YACHT ENGINE.

[illegible] By John W. Sullivan, New York.

are a single bronze casting, motion being obtained from the low pressure engine through the medium of light steel levers and links. The thrust bearing is arranged on the after end of the engine bed, where it is under the immediate supervision of the engineer. The space occupied by these engines fore-and-aft is one-third less than is required for the ordinary style of construction, while the entire engine can be oiled and inspected from the starting platform. On a recent trial of one of the small sizes of these engines some remarkable results were obtained. The diameters of the cylinders were 5, 8 and 13 inches respectively, with a common stroke of 8 inches. With a boiler pressure of 200 pounds, the number of revolutions made was 400, which is equal to a piston speed of 533 feet. Calculated percentage of admission, small cylinder 70, intermediate cylinder 60, large cylinder 70. Horse power estimated 38, 39½ and 40½, respectively, or 118 horse power altogether. The entire weight of engine including air and two feed pumps is 2,480 pounds. The propeller was 38 inch diameter by 6 feet pitch, and the speed attained was over 17 miles per hour, The vacuum maintained was 26 inches, which is very remarkable considering the high number of revolutions, while the engines were absolutely noiseless and developed no tendency to heat or cause trouble in any way. On the score of

economy these engines make a most satisfactory showing, furnishing a horse power for 1.61 pounds coal per hour, which is equivalent to less than one ton of coal for ten hours steaming, or a distance run of about 190 miles. The reputation gained by Mr. Sullivan in the manufacture of light yacht engines is an enviable one.

Fig. 81 shows a triple-expansion engine for steam yachts, by Messrs. Riley and Cowley of New York, who have met with great success with these engines. They have fitted them on board a large number of fine yachts, to the great satisfaction of all concerned.

It will be noticed that this engine presents a very open front, enabling attendant to adjust and inspect working parts at all times with facility, and allowing of repairs to be made without pulling engine apart. One of the marked features of these engines is the entire absence of vibration, even at highest speeds. This is due to the care with which the reciprocating parts are balanced. The ports and passages are so proportioned that excessive speeds are possible without undue frictional resistance to steam in passing through them. The engine being free from pumps, delivers its full power to the propeller, while the pumps being independent, may be adjusted to their work. It is self-contained, and of exceeding light weight.

FIG. 81—TRIPLE-EXPANSION ENGINE FOR STEAM YACHTS OF RILEY & COWLEY.

OPPOSITE PAGE [illegible].

CHAPTER XXIX.

TRIPLE-EXPANSION ENGINES, S. S. "COLUMBIA," 12,000 I. H. P. (ENGLISH.)

THE screws are revolved by two sets of engines, each set capable of developing 6,250 horse power. The cylinders are 40, 66 and 101 inches in diameter, stroke 66 inches. They are carried on extremely massive double-legged box columns. The bed plate and columns are unusually strong; in fact, it is not difficult to see that all the machinery is of dimensions very much in excess of that actually required. The wisdom of this was well illustrated when these enormous engines were running at full speed.

Steel is freely used in the moving parts of the machinery. The reversing gear is very rapid and noiseless in its action. The shafting is of steel, and is hollow. The crank shaft is 20½ inches diameter, the tunnel and propeller shafting being 19½ inches and 20½ inches in diameter respectively. The thrust-block is unusually large, and is of the adjustable open "horseshoe" patent. The glands in the engines are packed with metallic packing, and throughout the long trip, with the engines at full speed the whole time, all the glands not only remained tight, but

the piston and slide rods were as bright as possible, and no trouble was experienced on this head whatever.

The slides for the H. P. and I. P. cylinder are of the piston type, and in the L. P. cylinder the long D slide is fitted with a balance back. All the slide rods are balanced. The slide valves are worked by the ordinary link motion of the single bar type. Tail end piston rods for the cylinders are not fitted, but the shoes on the piston rod heads and the guide plate of the column are given very large surfaces. The pistons themselves are very deep. Condensers are placed at the back of the engines, and are of the usual surface pattern. The air pumps are driven by a rocking lever off the H and L cross heads. These are the only engines driven in connection with the main engines. Bilge feed and each circulating pump are driven by a pair of independent engines, made to a special design. There are two three-bladed steel propellers, their diameter being 18 feet and 32 feet pitch. The total blade area is 96 square feet, and the total disk area is 509 square feet. The bosses are of steel, and are 4 feet 6 inches in diameter.

There are 9 boilers, in three groups of three each, and each group, together with its coal supply, is placed in a separate water-tight compartment. Six of these boilers are 17 feet 3 inches long, and 15 feet 4 inches diameter, and

FIG. 82—TRIPLE-EXPANSION ENGINES OF THE S. S. COLUMBIA, 12

12,000 I. H. P. BY MESSRS. LAIRD BROS., BIRKENHEAD, ENGLAND.

OPPOSITE PAGE [illegible].

the remaining three are the same length, but 14 feet three inches in diameter. All the boilers are double-ended. The total heating surface is 35,000 square feet, and total grate area 1,220 square feet. The working pressure of steam is 150 pounds per square inch. Each group of boilers is supplied with a separate feed-pump and temperature compensator or feed heater. Feed injectors are also fitted to each boiler. The main steam pipes are so arranged that any one boiler or group of boilers can be used or shut off as the case may be. These engines are given here as an example of the best English practice.

CHAPTER XXX.

A COMPOUND MARINE ENGINE.

FIGS. 83 and 84 show a very remarkable set of compound engines lately completed for the Italian armor-clad ship, *Ruggiero di Laura*, at the shops of Maudslay, Sons & Field, of London, England, from the designs of Mr. Charles Sells, the head of the Engineering Department of that firm. The engines are of the three-cylinder compound type, having a high-pressure cylinder 61 in. in diameter and two low-pressure cylinders 80 in. in diameter each; the stroke of all being 39 in. As shown in Fig. 84, the high-pressure cylinder is set in the middle, with a low-pressure cylinder on each side. The engines are upright and act directly upon the shaft, the cranks being set 120° apart. The framework is entirely of steel and is thoroughly braced together, so as to secure the greatest possible rigidity, combined with lightness, and in this way it has been found possible to obtain great power with a very moderate weight. With the exception of the cylinders, steel and gun-metal are the only materials entering into the construction of the engines.

The valve-gear of these engines is of the Joy

pattern, fitted on the sling-link plan, in which the sliding block is replaced by an oscillating link. This type of gear is preferred for large

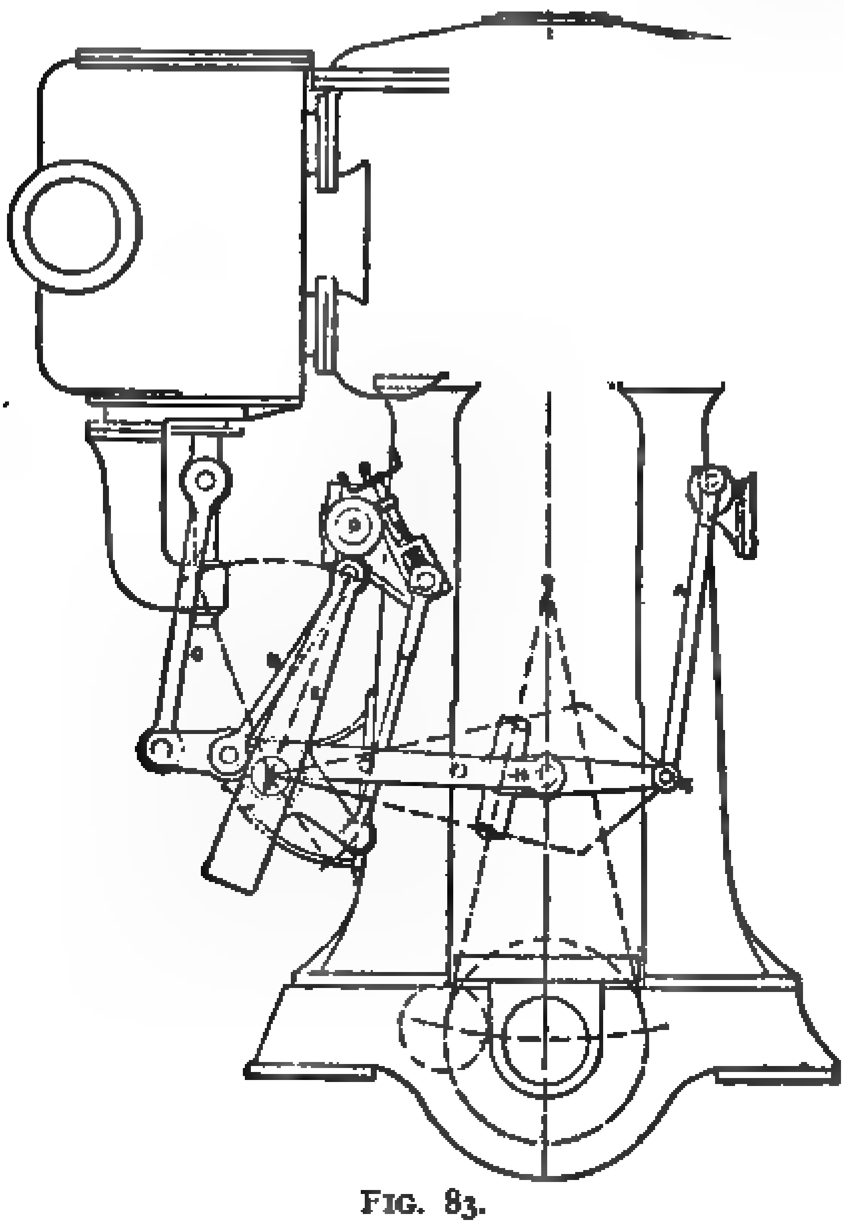

FIG. 83.

engines in preference to the ordinary Joy gear, which is considered more suitable for small en-

gines. An outline of the valve-gear is shown in the smaller cut, and its operation will be readily understood from this diagram. The sling-link *D* is suspended from a horseshoe lever *E*, which is supported in the fixed trunnion bearings *F*. The position of the horseshoe lever controls the cut-off and the direction of the motion, and this position is controlled by the main reversing lever *G*, a screw adjustment *B* being provided for the high-pressure cylinder, in order to give facilities for adjusting the proportion of cut-off in this and the two low-pressure cylinders. Motion is transmitted to the valve-lever *I* and the connecting-rod *C* by means of a connecting-lever *HK* suspended from the rod *A*. This arrangement of levers constitutes a kind of parallel motion and insures a correct cut-off both in forward and backward gear.

The contract for these engines provided that they should develop at least 10,000 H. P., but on the official trial trips, in the Gulf of Spezia, they attained a maximum of 12,000 indicated H. P., and the average for the whole trial was 11,400 indicated H. P. The engines thus showed a result of 14 per cent. in excess of the contract requirements.

FIG. 84. COMPOUND ENGINES OF ITAI

ITALIAN CRUISER "RUGGIERO DI LAURA."

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DESCRIPTION OF THE TRIPLE-EXPANSION ENGINES OF THE SOUTH AFRICAN MAIL STEAMER "DUNOTTAR CASTLE."

Built and Engined by the Fairfield Shipbuilding and Engineering Company, Limited, Glasgow.

These engines, (Fig. 85) which are of the usual Fairfield type, have three inverted cylinders and three cranks. The high-pressure cylinder is 38 inches in diameter, the medium-pressure cylinder 61½ inches, and the low-pressure cylinder 100 inches, each adapted for a piston stroke of 5ft. 6 in. The high-pressure cylinder is fitted with a piston-valve, and the medium and low-pressure cylinders are each fitted with a double-ported slide-valve, all of which are worked by double-eccentrics and link-motion. There is a special double-beat valve, with hand-gear for regulating the supply of steam to the engines, and a separate throttle-valve worked by the governor. The reversing of the engines is effected by one of Messrs. Brown Brothers & Co.'s steam and hydraulic reversing engines. The bilge-feed and air-pumps are driven off the crossheads of the high-pressure and low-pressure engines.

The crankshaft is in three pieces, each piece being built and interchangeable. This shaft, together with the thrust, tunnel, and propeller shafts, are forged of Siemens-Martin mild steel,

and have been supplied by Messrs. Vickers, Sons & Co., Limited, Sheffield. The diameter of the shaft is 18¼ inches, and of the propeller shaft 20¼ inches, both being hollow. The thrust block is of the horse-shoe type, with internal water-circulating arrangement, the shoes being hollow. The screw propeller has four blades of cast steel; the boss is also of cast steel, the whole being supplied by Messrs. Vickers, Sons & Co., Limited.

The water for condensing the steam is circulated through the surface condenser by two large centrifugal pumps, each with an independent engine and combined on one bed-plate.

The boilers for supplying steam to the engines are four in number, each being 15 ft. 3 in. in diameter and 18 ft. 8 in. long; they are multitubular, and fired from both ends, and arranged with two funnels. Each boiler has six Fox's corrugated furnaces, making a total of twenty-four furnaces for all the boilers, and each furnace is 3 ft. 8 in. in diameter. The boilers are constructed entirely of steel, and adapted for a working-pressure of 160 lbs. per square inch.

APPENDIX.

PROPERTIES OF SATURATED STEAM.

Pressure. By Steam Gauge.	Pressure. Total.	Temperature in Fahrenheit Degrees.	Volume. Compared with Water.	Volume. Cubic Feet of Steam from 1 lb. of Water.	Latent Heat in Fahrenheit Degrees.	Total Heat required to generate 1 lb. of Steam from Water at 32 deg. under constant pressure. In Heat Units.
0	15	212.0	1642	26.36	965.2	1146.1
5	20	228.0	1229	19.72	952.8	1150.9
10	25	240.1	996	15.99	945.3	1154.6
15	30	250.4	838	13.46	937.9	1157.8
20	35	259.3	726	11.65	931.6	1160.5
25	40	267.3	640	10.27	926.0	1162.9
30	45	274.4	572	9.18	920.9	1165.1
35	50	281.0	518	8.31	916.3	1167.1
40	55	287.1	474	7.61	912.0	1169.0
45	60	292.7	437	7.01	908.0	1170.7
50	65	298.0	405	6.49	904.2	1172.3
55	70	302.9	378	6.07	900.8	1173.8
60	75	307.5	353	5.68	897.5	1175.2
65	80	312.0	333	5.35	894.3	1176.5
70	85	316.1	314	5.05	891.4	1177.9
75	90	320.2	298	4.79	888.5	1179.1
80	95	324.1	283	4.55	885.8	1180.3
85	100	327.9	270	4.33	883.1	1181.4
90	105	331.3	257	4.14	880.7	1182.4
95	110	334.6	247	3.97	878.3	1183.5
100	115	338.0	237	3.80	875.9	1184.5
110	125	344.2	219	3.51	871.5	1186.4
120	135	350.1	203	3.27	867.4	1188.2
130	145	355.6	190	3.06	863.5	1189.9
140	155	361.0	179	2.87	859.7	1191.5
150	165	366.0	169	2.71	856.2	1192.9
160	175	370.8	159	2.56	852.9	1194.4
170	185	375.3	151	2.43	849.6	1195.8
180	195	379.7	144	2.31	846.5	1197.2

TABLE SHOWING THE DIAMETERS AND AREAS OF CIRCLES.

SIZE.	AREA.	SIZE.	AREA.	SIZE.	AREA.
⅛	0.0123	16	201.06	54	2290.2
¼	0.0491	½	213.82	55	2375.8
⅜	0.1104	17	226.98	56	2463.0
½	0.1963	½	240.53	57	2551.8
⅝	0.3068	18	254.47	58	2642.1
¾	0.4418	½	268.80	59	2734.0
⅞	0.6013	19	283.53	60	2827.4
1	0 7854	½	298.65	61	2922.5
⅛	0.9440	20	314.16	62	3019.1
¼	1.227	½	330.06	63	3117.2
⅜	1.485	21	346.36	64	3217.0
½	1.767	½	363.05	65	3318.3
⅝	2.074	22	380.13	66	3421.2
¾	2.405	½	397.61	67	3525.7
⅞	2.761	23	415.48	68	3631.7
2	3.142	½	433.73	69	3739.3
¼	3.976	24	452.39	70	3848.5
½	4.909	½	471.43	71	3959.2
¾	5.939	25	490.87	72	4071.5
3	7.068	26	530.93	73	4185.4
¼	8.296	27	572.56	74	4300.8
½	9.621	28	615.75	75	4417.9
¾	11.044	29	660.52	76	4536.5
4	12.566	30	706.86	77	4656.7
½	15.904	31	754.77	78	4778.4
5	19.635	32	804.25	79	4901.7
½	23.758	33	855.30	80	5026.6
6	28.274	34	907.92	81	5153.0
½	33.183	35	962.11	82	5281.0
7	38.484	36	1017.9	83	5410.6
½	44.179	37	1075.2	84	5541.8
8	50.265	38	1134.1	85	5674.5
½	56.745	39	1194.6	86	5808.8
9	63.617	40	1256.6	87	5944.7
½	70.822	41	1320.2	88	6082.1
10	78.54	42	1385.4	89	6221.1
½	86.59	43	1452.2	90	6361.7
11	95.03	44	1520.5	91	6503.9
½	103.87	45	1590.4	92	6647.6
12	113.10	46	1661.9	93	6792.9
½	122.72	47	1734.9	94	6939.8
13	132.73	48	1809.6	95	7088.2
½	143.14	49	1885.7	96	7238.2
14	153.94	50	1963.5	97	7389.8
½	165.13	51	2042.8	98	7543.0
15	176.71	52	2123.7	99	7697.7
½	188.69	53	2206.2	100	7854.0

Table of Squares, Cubes, Square and Cube Roots of Numbers.

Cubes.	Squares.	Number.	Square Roots.	Cube Roots.
1	1	1	1·0000000	1·0000000
8	4	2	1·4142136	1·2599210
27	9	3	1·7320508	1·4422496
64	16	4	2·0000000	1·5874011
125	25	5	2·2360680	1·7099759
216	36	6	2·4494897	1·8171216
343	49	7	2·6457513	1·9129312
512	64	8	2·8284271	2·0000000
729	81	9	3·0000000	2·0800837
1000	100	10	3·1622777	2·1544347
1331	121	11	3·3166248	2·2239801
1728	144	12	3·4641016	2·2894286
2197	169	13	3·6055513	2·3513347
2744	196	14	3·7416574	2·4101422
3375	225	15	3·8729833	2·4262121
4096	256	16	4·0000000	2·5198421
4913	289	17	4·1231056	2·6712816
5832	324	18	4·2426407	2·6207417
6859	361	19	4·3588989	2·6684016
8000	400	20	4·4721360	2·7144177
9261	441	21	4·5825757	2·7589243
10648	484	22	4·6904158	2·8020393
12167	529	23	4·7958315	2·8438670
13824	576	24	4·8989795	2·8844991
15625	625	25	5·0000000	2·9240177
17576	676	26	5·0990195	2·9624960
19683	729	27	5·1961524	3·0000000
21952	784	28	5·2915026	3·0365889
24389	841	29	5·3851648	3·0723168
27011	900	30	5·4772256	3·1072325
29791	961	31	5·5677644	3·1413806
32768	1024	32	5·6568542	3·1748021
35937	1089	33	5·7445626	3·2075343
39304	1156	34	5·8309519	3·2396118

Table of Squares, Cubes, &c.—Continued.

Cubes.	Squares.	Number	Square Roots.	Cube Roots.
42875	1225	35	5·9164798	3·2710663
46656	1296	36	6·0000000	3·3019272
50653	1369	37	6·0827625	3·3322218
54872	1444	38	6·1644140	3·3619754
59319	1521	39	6·2449980	3·3912114
64000	1600	40	6·3245553	3·4199519
68921	1681	41	6·4031242	3·4482172
74088	1764	42	6·4807407	3·4760266
79507	1849	43	6·5574385	3·5033981
85184	1936	44	6·6332496	3·5303483
91125	2025	45	6·7082039	3·5568933
97336	2116	46	6·7823300	3·5830479
103823	2209	47	6·8556546	3·6088261
110592	2304	48	6·9282032	3·6342411
117649	2401	49	7·0000000	3·6593057
125000	2500	50	7·0710678	3·6840314
132651	2601	51	7·1414284	3·7084298
140608	2704	52	7·2111026	3·7325111
148877	2809	53	7·2801099	3·7562850
157464	2916	54	7·3484692	3·7797631
166375	3025	55	7·4161985	3·8029525
175616	3136	56	7·4833148	3·8258624
185193	3249	57	7·5498344	3·8485011
195012	3364	58	7·6157731	3·8708766
205379	3481	59	7·6811457	3·8929965
216000	3600	60	7·7459667	3·9147632
226981	3721	61	7·8102497	3·9304972
238328	3844	62	7·8740079	3·9578915
250047	3969	63	7·9372539	3·9790571
262144	4096	64	8·0000000	4·0000000
274625	4225	65	8·0622577	4·0207256
287496	4356	66	8·1240384	4·0412401
300763	4489	67	8·1853528	4·0615480
314432	4624	68	8·2462113	4·0816551

Table of Squares, Cubes, &c.—Continued.

Cubes.	Squares.	Number.	Square Roots.	Cube Roots.
328509	4761	69	8·3066239	4·1015661
343000	4900	70	8·3666003	4·1212853
357911	5041	71	8·4261498	4·1408178
373248	5184	72	8·4852814	4·1601676
389017	5329	73	8·5440037	4·1793390
405224	5476	74	8·6023253	4·1983364
421875	5625	75	8·6602540	4·2171633
438976	5776	76	8·7177979	4·2358286
456533	5929	77	8·7749644	4·2543210
474552	6084	78	8·8317609	4·2726586
493039	6241	79	8·8881944	4·2908404
512000	6400	80	8·9442799	4·3088695
531441	6561	81	9·0000000	4·3267487
551368	6724	82	9·0553851	4·3444815
571787	6889	83	9·1104336	4·3620707
592704	7056	84	9·1631514	4·3795191
614125	7225	85	9·2195445	4·3968296
636056	7396	86	9·2736185	4·4140049
658503	7569	87	9·3273791	4·4310476
681472	7744	88	9·3808315	4·4470692
704969	7921	89	9·4339811	4·4647451
729000	8100	90	9·4868330	4·4814047
753571	8281	91	9·5393920	4·4979414
778688	8464	92	9·5916630	4·5143574
804357	8649	93	9·6436508	4·5306549
830584	8836	94	9·6953597	4·5468359
857374	9025	95	9·7467943	4·5629026
884736	9216	96	9·7979590	4·5788570
912073	9409	97	9·8488578	4·5943009
949912	9604	98	9·8994949	4·6104363
970299	9801	99	9·9498744	4·6260650
1000000	10000	100	10·0000000	4·6415888
1030301	10201	101	10·0498756	4·6570095
1061228	10404	102	10·0995049	4·6723287

Table of Squares, Cubes, &c.—**Continued.**

Cubes.	Squares.	Number	Square Roots.	Cube Roots.
1092727	10609	103	10·1488916	4·6875482
1124864	10816	104	10·1980390	4·7026694
1157625	11025	105	10·2469503	4·7176940
1191016	11236	106	10·2956301	4·7326235
1225043	11449	107	10·3440804	4·7474594
1259712	11664	108	10·3923048	4·7622032
1295029	11881	109	10·4403065	4·7768562
1331000	12100	110	10·4880885	4·7914199
1367631	12321	111	10·5356538	4·8058995
1404928	12524	112	10·5830052	4·8202845
1442897	12769	113	10·6301458	4·8343881
1481344	12996	114	10·6770783	4·8488076
1520875	13225	115	10·7238053	4·8629442
1560896	13456	116	10·7703296	4·8769900
1601613	13689	117	10·8166538	4·8909732
1643032	13924	118	10·8627805	4·9048681
1685159	14161	119	10·9087121	4·9186847
1728000	14400	120	10·9544512	4·9324242
1771561	14641	121	11·0000000	4·9460874
1815848	14834	122	11·0453610	4·9596757
1860867	15129	123	11·0905365	4·9731898
1906624	15376	124	11·1355287	4·9866310
1953125	15625	125	11·1803399	5·0000000
2000376	15876	126	11·2249722	5·0132979
2048383	16129	127	11·2694277	5·0265257
2097152	16384	128	11·3137085	5·0396842
2146689	16641	129	11·3578167	5·0527743
2197000	16900	130	11·4017543	5·0657970
2248091	17161	131	11·4455231	5·0787531
2299968	17424	132	11·4891253	5·0916434
2352637	17689	133	11·5325626	5·1044687
2408104	17956	134	11·5758369	5·1172259
2460375	18225	135	11·6189500	5·1299278
2515456	18496	136	11·6619038	5·1425632

Table of Squares, Cubes, &c.—Continued.

Cubes.	Squares.	Number.	Square Roots.	Cube Roots.
2571353	18769	137	11·7046999	5·1550367
2628072	19044	138	11·7473444	5·1676493
2685619	19321	139	11·7898261	5·1801015
2744000	19600	140	11·8321596	5·1924941
2803221	19881	141	11·8743421	5·2048279
2863288	20164	142	11·9163753	5·2171034
2924207	20449	143	11·9582607	5·2293315
2985984	20736	144	12·0000000	5·2414828
3048625	21025	145	12·0415946	5·2535879
3112136	21316	146	12·0830460	5·2656374
3176523	21609	147	12·1243557	5·2776321
3241792	21904	148	12·1655251	5·2895725
3307949	22211	149	12·2065556	5·3014592
3375000	22500	150	12·2474487	5·3132928
3442951	22801	151	12·2882057	5·3250740
3511008	23104	152	12·3288280	5·3368033
3581577	23409	153	12·3693169	5·3484812
3652264	23716	154	12·4096736	5·3601084
3723875	24025	155	12·4498996	5·3716854
3796416	24336	156	12·4899960	5·3832126
3869893	24649	157	12·5299641	5·3946907
3944312	24964	158	12·5698051	5·4061202
4019679	25281	159	12·6095202	5·4178015
4096000	25600	160	12·6491106	5·4258352
4173281	25921	161	12·6885775	5·4401218
4251528	26244	162	12·7279221	5·4513618
4330747	26569	163	12·7671453	5·4625556
4410944	26896	164	12·8062485	5·4737037
4492125	27225	165	12·8432326	5·4848066
4574296	27556	166	12·8840987	5·4958647
4657463	27889	167	12·9228480	5·5068784
4741632	28224	168	12·9614814	5·5178484
4826809	28561	169	13·0000000	5·5287748
4913000	28900	170	13·0384048	5·5396583

Table of Squares, Cubes, &c.—Continued.

Cubes.	Squares.	Number.	Square Roots.	Cube Roots.
5000211	29241	171	13·0766968	5·5504990
5088448	29584	172	13·1148770	5·5612578
5177717	29929	173	13·1529464	5·5720546
5268024	30276	174	13·1909060	5·5827702
5359375	30625	175	13·2287566	5·5934447
5451776	30976	176	13·2664992	5·6040787
5545233	31329	177	13·3041347	5·6146724
5639752	31684	178	13·3416641	5·6252263
5735339	32041	179	13·3790882	5·6357408
5832000	32400	180	13·4164079	5·6462162
5929741	32761	181	13·4536240	5·6566528
6028568	33124	182	13·4907376	5·6670511
6128487	33489	183	13·5277493	5·6774114
6229504	33856	184	13·5646600	5·6877340
6331625	34225	185	13·6014705	5·6980192
6434856	34596	186	13·6381817	5·7082675
6539203	34969	187	13·6747943	5·7184791
6644672	35344	188	13·7113092	5·7286543
6751260	35721	189	13·7477271	5·7387936
6859000	36100	190	13·7840488	5·7488971
6967871	36481	191	13·8202750	5·7589652
7077888	36864	192	13·8564065	5·7689982
7189517	37249	193	13·8924400	5·7789966
7301384	37636	194	13·9283883	5·7889604
7414875	38025	195	13·9642400	5·7988900
7529536	38416	196	14·0000000	5·8087857
7645373	38809	197	14·0356688	5·8186479
7762392	39204	198	14·0712473	5·8284867
7880599	39601	199	14·1067360	5·8382725
8000000	40000	200	14·1421356	5·8480355

Table of Approximate Numbers, for various purposes.

Diameter of a circle	×	3·1416	= the circumference.
Circumference "	×	·31831	= the diameter.
Diameter "	×	·8862	= the side of an equal square.
Side of a square......	×	1·128	= the diameter of an equal circle.
Square of diameter	×	·7854	= the area of a circle.
Square root of area	×	1·12837	= the diameter of equal circle.
Square of the diameter of a sphere	×	3·1416	= convex surface.
Cube of ditto	×	·5236	= solidity.
Diameter of a sphere	×	·806	= dimensions of equal cube.
Diameter of a sphere	×	·6667	= length of equal cylinder.
Square inches.........	×	·00695	= square feet.
Cubic inches...........	×	·00058	= cubic feet.
Cubic feet..............	×	·03704	= cubic yards
Cylindrical inches...	×	·0004546	= cubic feet.
Cylindrical feet	×	·02909	= cubic yards.
Cubic inches...........	×	·003607	= imperial gallons.
Cubic feet..............	×	·6232	= "
Cylindrical inches...	×	·002832	= "
Cylindrical feet	×	4·895	= "
183·346 circular inches................			= 1 square foot.
2200 cylindrical inches................			= 1 cubic foot.
Avoirdupois pounds	×	·009	= cwts.
Avoirdupois pounds	×	·00045	= tons.
Lineal feet.............	×	·00019	= English statute miles.
Lineal yards...........	×	·000568	= "

INDEX.

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